HEATING & VENTILATING

For Engineers and Contractors Controlling the Design, Specification, Installation and Maintenance of Heating, Ventilating and Air Conditioning

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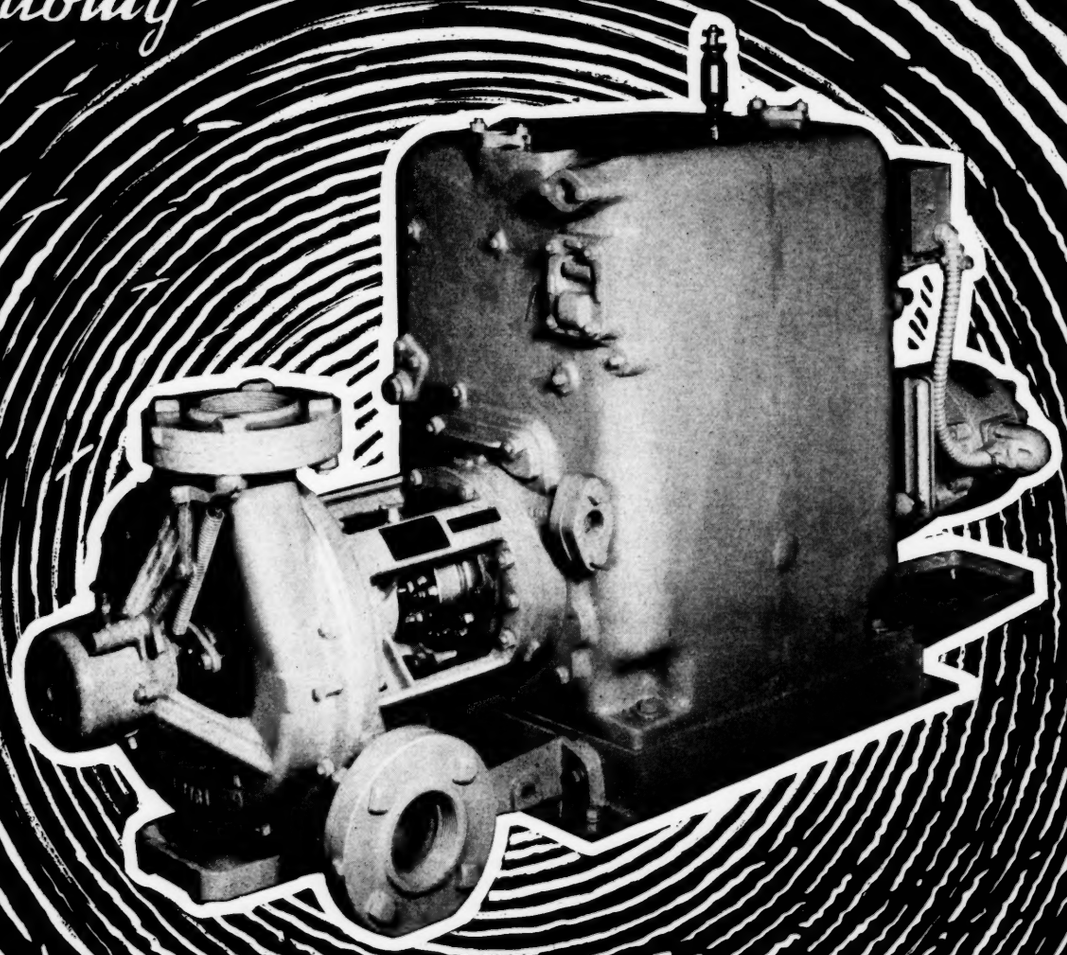

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A Diesel Engine for Heating?

By JOSEPH F. KERN, Jr.†

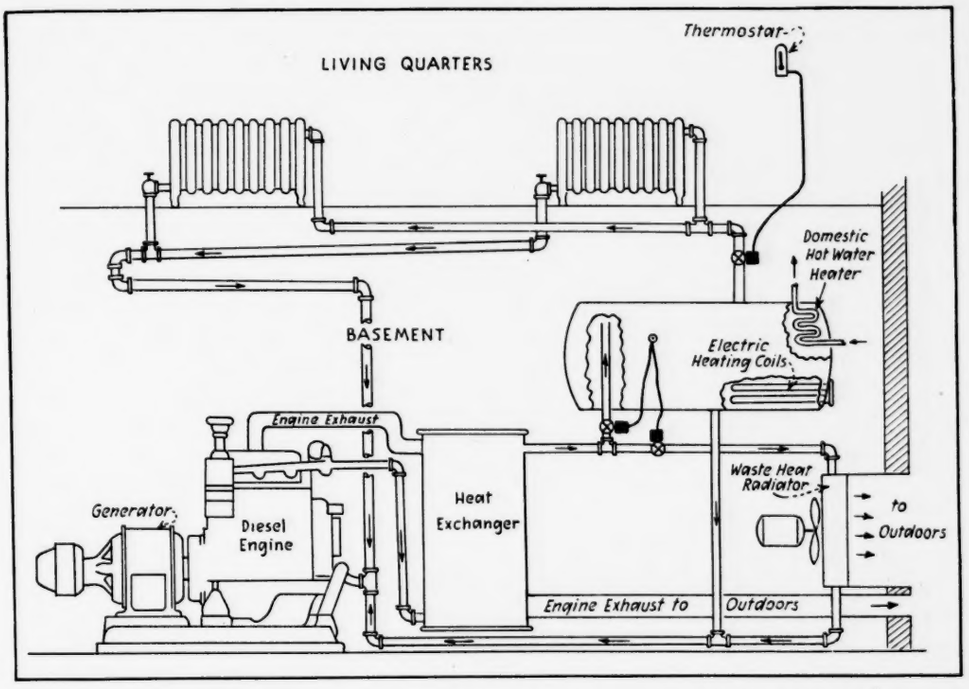

Fig. 1. Proposed arrangement of equipment when a diesel engine is used for heating.

THE utilization of the waste heat from a diesel engine for heating buildings is a fairly common practice at the present time, especially in industrial plants where large amounts of electricity are produced for manufacturing operations. From time to time, however, the suggestion has been made that, since such a large amount of the heat put into a diesel engine can be recovered, it might prove practical to use such a device for heating and at the same time secure sufficient power as a byproduct to furnish enough electricity to meet the needs of the building. In other words the proposal suggests the converse of present practice—the installation of a diesel primarily to produce heat with the generation of power only as a byproduct.

Briefly, one suggested method of operation is as follows: an internal combustion engine is direct connected to a generator and the heat in the exhaust gases and cooling water together with some of the electricity is used to heat the building. The remaining electricity could be used for lighting and cooking.

It is stated by some advocates of this method that the operating cost would be less than the cost of heating with an oil burner alone and at the same time electricity would be furnished practically free. It would make the small building owner independent of utilities and of breakdowns in power lines during storms. It is also claimed that the unit could produce power at such a low rate even during the summer that it would be possible to cool a building completely at an extremely low cost.

Several disadvantages, such as high initial cost and the possibility of mechanical troubles, are apparent on only a superficial examination of the idea. Whether or not these disadvantages would make the idea impractical can be determined only after a thorough investigation and tests on the actual equipment. The most we can hope to do in the limited space available is to attempt to get a clear idea of the method of operation and to point out some of the more obvious characteristics of the plan.

A diagram of one method of hooking up the apparatus for heating a residence is shown in Fig. 1. Other methods are possible but for the purpose of illustration let us assume that this is the one used. Cooling water for the diesel engine is obtained either from the return of the hot water heating system or from the waste heat radiator. After leaving the engine, the water flows through a heat exchanger where it absorbs heat from the exhaust gases. A large storage tank is used to hold the hot water until it is used in the heating system. If there is an excess of hot water it passes through the waste heat exchanger where it is cooled sufficiently to allow it to be used again for cooling the engine. Domestic hot water is obtained by passing cold water through a submerged coil in the storage tank. A control valve in the heating riser from the storage tank is actuated by a thermostat in the space to be heated.

As soon as there is a demand for heat, the engine

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and generator start. Electricity is generated in an amount equal to the heat load less the amount of waste heat recovered from the exhaust gases and cooling water. Any demand for electricity for cooking or lighting at the same time increases the load on the engine but less electricity is then used for heating since there is a greater supply of waste heat available to supply the heat demand. In other words, the heat demand of the system is always satisfied as far as possible by the waste heat and only when this supply is insufficient is electricity generated to make up the difference. In case there is no demand for heat when electricity is needed, hot water will be stored in the large tank until the water reaches a predetermined temperature. After this, the hot water from the exhaust gas heater will pass through a radiator type cooler which removes the heat from the water and carries it outdoors.

During the summer most of the heat would be exhausted in this manner unless some means of utilizing it were found. One such means would be dehumidifying equipment using heat as a motivating force. The heat from the exhaust gases and electrical heating coils could be used to control the humidity while an electrical refrigerating machine could be used for air cooling.

Now that the method of operation has been outlined let us examine the individual pieces of equipment.

The diesel engine is simply an internal combustion engine using liquid fuel injected directly into the cylinder and depending upon the heat of compression for ignition. The heat of combustion raises the pressure of the air in the cylinder, forces the piston down and thus rotates the crankshaft. The operation differs from the ordinary gasoline engine in that the mixture of fuel and air is not compressed and then ignited by an electrical spark. In the diesel engine the air alone is compressed to a high pressure and then the liquid fuel is injected into the cylinder. The high temperature of the compressed air immediately causes the liquid fuel to ignite.

The primary reason for selecting a diesel engine instead of some other type of internal combustion engine is the fact that it has the ability to burn cheap types of oils which compare favorably in cost with the oils used for building heating.

Moreover, there are a large number of commercial diesel engines which with a few minor changes are considered suitable for this type of operation. Large numbers of these diesels have been installed in the past few years to generate power in isolated localities and in buildings where an independent source of power is desired.

Practically any size engine from 3 hp. up can be obtained and while it is true that most of the smaller engines sold are equipped for hand starting it is possible to secure those which can be started automatically with either compressed air or electricity. In general most of the small engines are of the four cycle rather than the two cycle type. From the standpoint of using a diesel for heating, the four cycle type is more desirable because it has higher exhaust temperatures which makes it possible to secure a more efficient recovery of heat from the exhaust gases.

The heat exchanger is simply a device for transferring the heat from the exhaust gases to the water used for heating. It is possible to use an ordinary house-heating boiler as a waste heat exchanger if a high efficiency is not desired. The waste heat exchangers manufactured for this purpose have a much greater amount of heating surface than the usual boiler so that it is possible to recover a large amount of heat from the relatively low temperature exhaust gases.

Exhaust Gases

The exhaust gases discharged from the engine are at a temperature in the neighborhood of 700° to 800°. It has not been found practical to cool these gases below 300° because of the possibility of condensing the water vapor. Condensation of the water vapor will do much damage to the heat transfer surface if large amounts of acids are present in the exhaust gases.

The amount of gas discharged by the engine is dependent on two factors—the amount of fuel being consumed and the air-fuel ratio. The minimum amount of air needed to secure complete combustion of 1 lb. of fuel oil is 14.2 lb. In practice the amount of air used is always greater than this because it is impossible with this amount to get a proper mixture of the air and oil and some of the oil will pass through the engine unburned. Somewhere between 1¼ and 2 times the minimum amount is used in modern diesels.

In order to determine the amount of heat recoverable from the exhaust gases it is necessary to determine the amount of gas produced per pound of oil consumed. If we assume that 24 lb. of air are required to burn 1 lb. of oil, the amount of exhaust gases will be 24 + 1 or 25 lb. The amount of heat recovered per pound of fuel in cooling the gas from 700° to 300° is 25 lb. × 0.25 × (700° — 300°) = 2,500 B.t.u. where 0.25 is the specific heat of the exhaust gas.

Only at full load, however, is the exhaust gas discharged at 700° for as the load decreases the exhaust temperature decreases. When the engine has no load on it the exhaust temperature will be close to 300° so that practically no heat can be recovered.

Cooling Water

In most small diesels the cooling water leaves the engine at a temperature somewhere in the neighborhood of 120° and 140°. The principal reason for the use of such low temperatures is to prevent the formation of scale in the water cooling chambers of the engine. If, however, a very pure water is used or if the water is used over and over it is possible and even desirable to have the cooling water discharged at higher temperatures, for many engineers state that the engine becomes more efficient at the higher temperatures. With the arrangement of apparatus as shown, the temperature of the water leaving the engine will be somewhere between 150° to 180° when the entering temperature is between 120° and 160°.

It is possible to recover all of the heat in the cooling water. From actual tests on diesel engines it has been found that about 2,500 B.t.u. per hp. is removed by the cooling water every hour. This is for full load —at decreased loads the figure increases slightly since more fuel is used per horsepower.

Sizing the Engine

The size of the engine would be determined by the total energy load of the building. For the purpose of illustration let us assume that the system is to be installed in a residence housing five people and having a heat requirement of 70,000 B.t.u. per hr. when 70° is maintained in 0° weather. The system should also furnish sufficient electricity for the usual needs of such a building.

Since the size of the unit is determined by the maximum load and since the load would be at a maximum during a severe winter day it will first be necessary to get an idea of the magnitude of this load. For the purpose of this example the hourly heat requirements have been selected as shown in Fig. 2. From available data on the consumption of hot water and electricity in residences the other figures on this chart have been selected.

The space below the top solid line represents the total load on the unit in B.t.u. per hour, while the shaded space shows the heating load, the dotted space the hot water load, and the solid space the electrical load. All loads are in B.t.u. per hour.

The maximum load on the system as shown by Fig. 2 would occur between 7 and 8 a.m. on a severe winter day and would be about 81,000 B.t.u. per hr. Since this demand is only for a short period and falls to about 71,000 B.t.u. per hr. between 8 and 9 it would not be necessary to have an engine with this output. The use of a large hot water storage tank will carry the system over these peak periods safely.

Small diesel engines will consume between 0.40 and 0.50 lb. of fuel per hp. when operating at full load. Thus, when oil with 19,000 B.t.u. per lb. is used, the heat input will be between 7,600 and 9,500 B.t.u. per hp. If we assume the engine burns 0.46 lb., the heat input is 8,750 B.t.u. per hp.

For the purpose of estimating the size of the engine we can assume that all of the heat input to the engine can be recovered except about 60% of the heat in the exhaust gas. Since about 30% of the total heat input appears in the exhaust gas, about 18% of the heat input is wasted. In other words about 7,200 B.t.u. (8,750 $\times$ 0.82) can be recovered out of every 8,750 B.t.u. of heat put in.

Thus, if an engine were selected which could meet the maximum heat requirement of 81,000 B.t.u. per hr., it would need a capacity of 81,000 B.t.u. $\div$ 7,200 B.t.u. per hp., or 11.1 hp. Since this peak load is reached only for a short period a 10-hp. engine could be used.

Heat Balance

A heat balance diagram of a 10-hp. diesel engine based on the figures presented previously is shown in Fig. 3. In this chart the space below the heavy top line represents the total heat input of the engine at

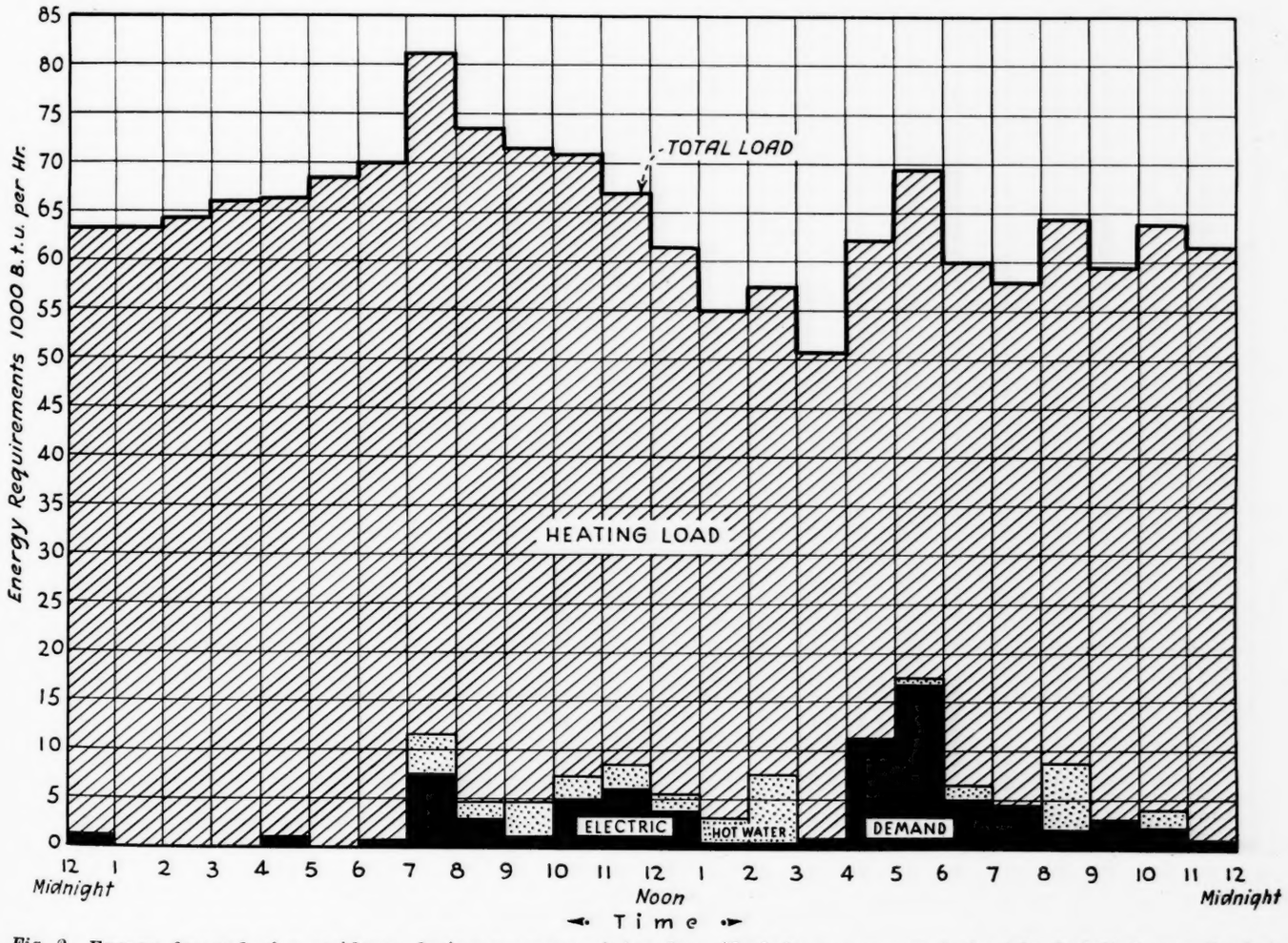


Fig. 2. Energy demand of a residence during a severe winter day. Shaded area represents heat load, dotted area the hot water load, and solid area the electric demand.

various loads. The space marked waste heat represents the amount of heat in the exhaust gases which cannot be recovered because it is at temperatures below 300°. All of the space below the line between the spaces marked waste heat and radiation represents the amount of heat which can be recovered. The reason why the heat radiated from the engine is included in the recoverable heat is because it goes to heat the basement and so reduces the heat load on the building.

Efficiency of the System

From the chart of the heat balance of the engine it can be seen that the expected efficiency of the engine when operating as a heat generating device at full load is 72,500 ÷ 88,000 B.t.u. or 82.5%. This figure is believed to be fairly representative of efficiencies reached by engines in actual operation. The average oil burner when in good adjustment may also have an efficiency of about 82% assuming that boiler radiation is not a loss since it goes to heat the basement. In other words the oil burner when in good adjustment should have an overall efficiency equal to that of a diesel engine operating as a heat producer. This is to be expected because both are merely devices for burning oil and if the exhaust gases are discharged in about the same condition they should have about the same efficiencies.

From the foregoing it can be seen that if there is a demand for electricity the diesel engine has an advantage in that it produces some of its energy in the form of electricity.

Cost

Probably the main objection to the use of a diesel engine as a heat producer is high initial cost. One manufacturer states that a diesel generator set will cost about $100 per kw-hr. output and the installation cost about $300 to $500. Thus a 10-hp. or 7½-kw. set would cost about $750 plus $300 for installation. Added to this would be about $200 for the waste heat boiler, $200 for the storage tank, $100 for the waste heat radiator, and $100 for the controls. The total cost would be about $1,650 against about $500 for an oil burner. On the other hand, if a diesel was used for heating, some operating saving could be credited since it would not be necessary to buy electricity for lighting and cooking.

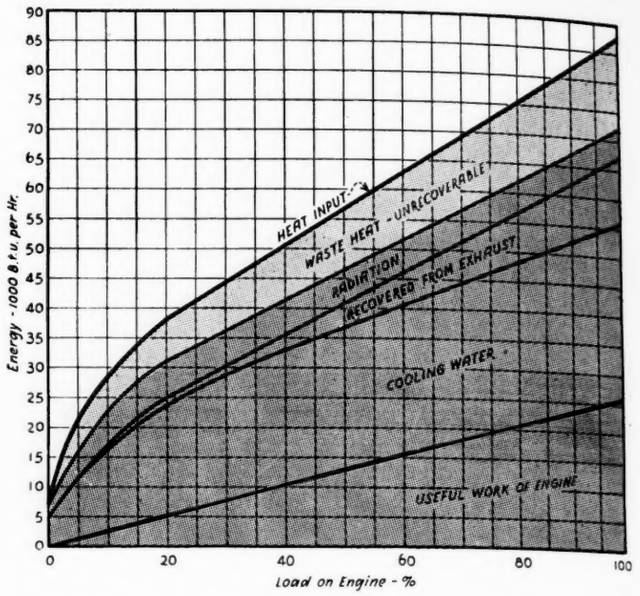


Fig. 3. Heat balance of 10-hp. diesel engine. Shaded area represents total heat input. Dark area is the heat recovered and light area the heat wasted.

Stoker Saves Paint Manufacturer 20% in Fuel

INDUSTRIAL modernization might well begin in the boiler room, judging from the satisfactory experience of the DeVoe & Reynolds Co., Brooklyn, N. Y., paint manufacturer, with the mechanical stoker it had installed a year ago. The Public Service Heat & Power Co., Jamaica, L. I., which made the sale, traces upwards of 20% fuel saving, this being corroborated by J. D. Grayson, an executive of the paint company.

Prior to the stoker installation, run-of-mine bituminous coal was fired by hand in two 150-hp. tubular brick set boilers. Combustion of 11 lb. per sq. ft. of grate was the rule and boilers operated at about 75% rating. Following the installation of the stoker it was found that fuel usage often reached 32 lb. per sq. ft. of grate and boilers could be stepped up to 150% rating. A short trial demonstrated that only one boiler need be operated and this has been the case since the modernization in February, 1935.

The second boiler now stands as an emergency unit, although conditions thus far have not required such assistance. The function of the spare at present is as an incinerator. Around the corner, but outside the present boiler room, there was formerly a third boiler kept as a standby. This unit recently was disposed of and the space is now used for materials storage.

Fuel was unchanged, continuing to be run of mine. Tonnage used, however, has been considerably less than with hand firing. The gross reduction in tonnage over the period from February through September of 1935, versus the same period in 1934, was 30%. This figure should be modified because the number of degree-days during the period was 13% less this year. It is, nevertheless, agreed that net tonnage saving approached 20%.

Not all the credit for the fuel economy should be attributed to the stoker, though, because other improvements were made which are known to have saved fuel. For instance, some dozen or so swinging doors were installed in places where no doors between plant rooms existed previously. Also an open elevator shaft was enclosed. It is difficult to measure the benefits thus derived in terms of fuel dollars saved but the factory people assert that the doors have contributed.

Steam Generation by Burning Wood

By R. E. SUMMERS†

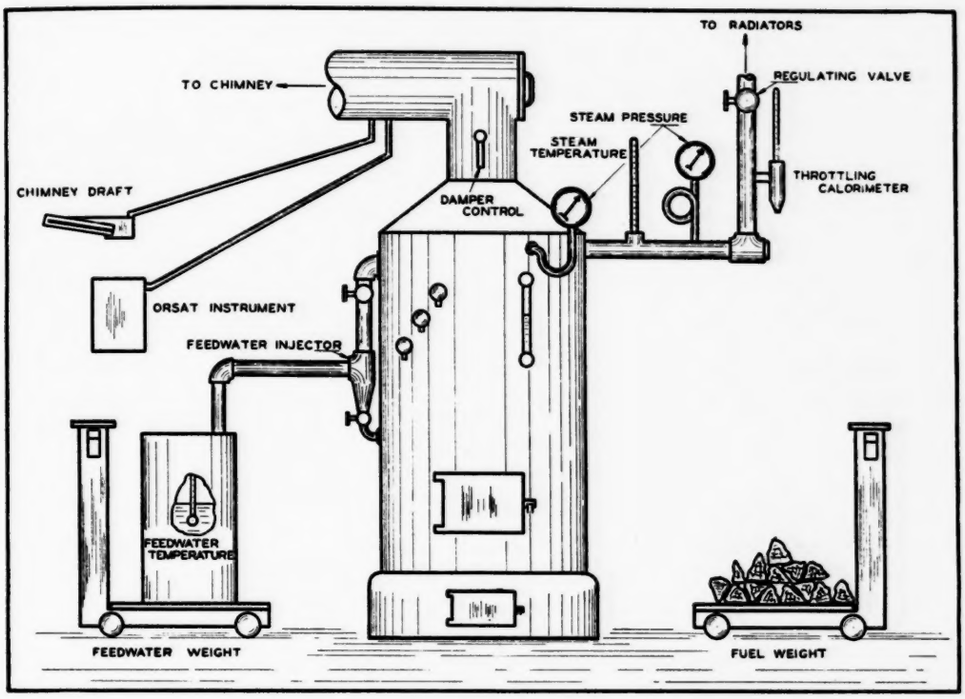


Fig. 1. Setup for testing boilers fired with wood.

DESPITE the long and widespread use of the vertical fire-tube boiler, little test data are available in engineering literature for boilers of that type. The lack of definite performance figures is especially pronounced when wood is the fuel used. That boiler has usually been the possession of the man of small steam needs and limited operating interest. Wood fired, it has especially come under the care of the casual and inexperienced operator. Thus, a type of boiler most likely to be used by an owner least able to make scientific tests or determinations is singularly without information on efficiency and capacity.

Results are presented here for a number of efficiency tests of a small vertical fire-tube boiler operated at various known percentages of rated capacity. The boiler tested was of 26-in. diameter, and of simple cylindrical form. It contained 37 2-in. tubes each 36.5 in. long. Water heating surface, below normal water level at the center try cock, totaled 42.7 sq. ft.; superheating surface, above normal water level, 21.0 sq. ft. No insulation was applied to the boiler. A common wood grate of the stationary hand-fired type provided a surface of 2.29 sq. ft. The firebox had a diameter of 21.5 in. and a height of 19.6 in. Natural draft was provided by an oversize brick chimney 45 ft. high above the top of the boiler. A simple butterfly valve damper fitted in the chimney connection was partly closed at low steam outputs. Air for combustion was normally all admitted through an ashpit door that was under hand adjustment.

†Assistant Professor of Mechanical Engineering, Oregon State College.

Fuel was several different lots of second-growth Douglas fir cordwood sawed to 12-in. lengths and split to suit, often to cook-stove size. Firing was intermittent at approximate 5-min. intervals. All fuel was carefully sampled, tested, and weighed. Feedwater was weighed cold tap water fed to the boiler by an injector operated once in each 10 min. of test time. Steam was released from the boiler continuously during the test, the boiler pressure being maintained near 60 lb. per sq. in. gage by constant manual regulation of the steam stop valve. Disposal of the steam was by condensation in a building heating system. A throttling calorimeter fitted to the steam outlet pipe was operated, as necessary, to verify steam conditions when low superheats were encountered, as at some loads. A diagram of apparatus is shown in Fig. 1.

All test runs made at any approximately fixed rate of steam output had a duration of 10 hr., subsequent to attainment of a steady operating state. Runs were made at widely different times, over a period of years, and in the order of designation. The results of any one trial were used in determining a single efficiency result. Data obtained represent, therefore, the product of tests consuming an aggregate testing time of more than an entire week. The long duration of trial at any given boiler output was chosen to insure freedom from drift in important valves during any run, as well as to minimize the effect of starting and stopping errors. Those errors were, further, reduced to the minimum through correction of fuel and feedwater quantities of any one trial for differences in starting and stopping

TABLE 1. SUMMARY OF TEST RESULTS

Trial designation	A	B	C	D	E	F
Trial duration, time, hr.	10.00	10.00	10.00	10.00	10.00	10.00
Steam pressure, avg., lb. per sq. in. gage	59.8	59.2	60.2	60.2	60.0	61.9
Barometric pressure, avg., lb. per sq. in. absolute	14.8	14.6	14.5	14.7	14.6	14.8
Chimney draft, avg., in. of water	—	—	0.18	0.19	0.18	0.17
Feedwater temperature, avg., degrees F.	62.0	75.5	62.5	70.4	82.7	77.7
Total steam temperature, avg., degrees F.	339.2	334.9	322.2	311.2	311.8	326.4
Boiler room temperature, avg., degrees F.	77.4	81.2	79.6	82.1	69.9	75.5
CO_2 in flue gas, avg., per cent by volume	—	11.1	—	9.3	—	8.0
Total feedwater, corrected for starting and stopping, lb.	2,934	3,653	4,120	3,329	1,921	2,510
Total fuel, corrected for starting and stopping, lb.	1,279	1,510	1,473	1,174	763	799
Heating value of dry fuel, B.t.u. in 1 lb.	9,063	9,063	9,063	9,063	9,063	9,063
Moisture in fuel as fired, per cent by weight	28.4	16.6	11.3	16.1	16.5	11.9
Overall efficiency; boiler, furnace, and grate; per cent	41.4	36.9	40.3	42.8	37.7	45.2
Output, per cent manufacturer's rating	240	295	334	267	152	201

levels of both fuel bed and water surface. A summary of results for all tests is presented in Table 1. Fig. 2 shows the variation of overall efficiency of the boiler, furnace, and grate with boiler output expressed as a percentage of manufacturer's rating.

Significant facts are brought out by the efficiency and capacity results. The absolute efficiency values, with a maximum around 45%, are not greatly different from those expected of similar furnace-boiler installations as, for instance, large, wood-fired, domestic boilers and stoves. Efficiency is seen to vary as a function of output much as for the larger power-boiler installations. It is interesting, moreover, that, as is usual with power plant boilers, maximum efficiencies are attained with steam outputs well in excess of 100% of the rated boiler capacity.

The higher outputs obtained exceed those frequently thought to be maximum for boilers of this type. Their attainment is perhaps not to be doubted when steam generating capacities of the smallest vertical fire-tube boilers are viewed as a function of the grate surface provision. The boiler tested had 1 sq. ft. of grate surface for 18.6 sq. ft. of water heating surface. An allowance of but 1 sq. ft. of grate for as much as 35 sq. ft. of water heating surface is not unusual for some water-tube types of power plant boilers fired with wood-waste fuels. Split cordwood used for these tests probably requires a lesser grate provision than sawdust or hogged wood. It should be remembered, of course, that the vertical fire-tube boiler tested was a very small one and that the ratio of grate to heating surface becomes less favorable with increase in boiler size. The insufficient provision of grate in the larger sizes largely restricts the vertical fire-tube boiler to small horsepowers or dictates construction as in the Manning boiler. The excellent draft provided during these tests accounts, in part, for the obtained high percentages of rated capacity. Carbon dioxide percentages from flue gas analyses for a few of the tests appear somewhat inconsistent and are not entirely reliable. Difficulty of duplicating firing conditions between tests, made by different crews, was obvious. The high moisture content of wood used for trial A, compared with fuel moisture percentages for other tests, is a logical result of testing conditions. Test A was made in the late winter season; all other tests were made in the spring or early summer, each in a different year.

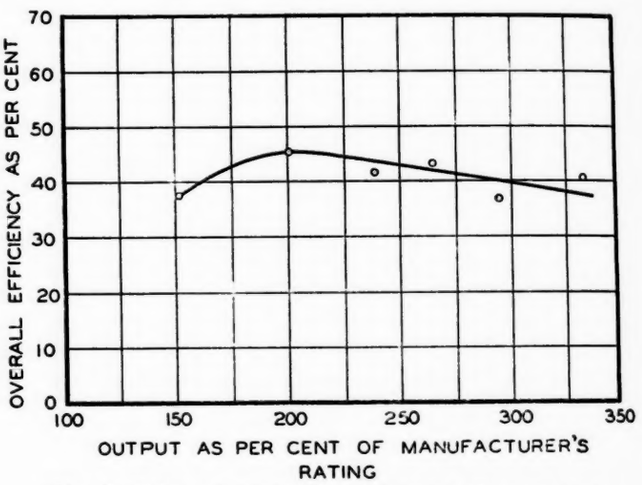


Fig. 2. Effect of "forcing" wood-fired boilers on their overall efficiencies.

How to Make Surveys for Stoker Installations

By NOEL CUNNINGHAM

In surveying a heating plant for a possible stoker installation we are concerned with three items: (1) the heater, because the amount of coal that can be effectively burned depends on the size of the firebox and indirect heating surface; (2) the load, which is made up of the heating load proper and that of service hot water; and (3) the stoker required, especially its proper size.

The Heater

All heaters are designed primarily to absorb into a heating medium the heat released by combustion of the fuel. The heat absorbing surfaces are of two kinds: direct—those which "see" the incandescent fuel and flame; and indirect—those over which the hot gas passes. Ordinarily, direct surface is about six times as effective per unit of area as is indirect surface.

Cast iron rectangular (or sectional) steam or hot water boilers, or round boilers with one intermediate section, when stoker fired, can absorb about 60% of the heat released in burning 5 lb. of coal per hr. per sq. ft. of horizontal, internal cross-sectional area measured at the firedoor. In round boilers with no intermediate section, 4 lb. per hr. per sq. ft. is maximum for stoker firing. In these boilers each intermediate section adds approximately ½ lb. per hr. per sq. ft. of area to the safe burning rate.

Cast iron warm air furnaces, stoker fired, will utilize about 60% of the heat released in burning 4 lb. of coal per hr. per sq. ft. of cross section[1]; steel furnaces with rectangular fireboxes, on forced circulation and stoker equipped, will handle as much as 8 lb. of coal per hr. per sq. ft.

Some of the heat released in a boiler or furnace when coal is burned is partly dissipated by radiation from outer heater surfaces (about 4% from heaters properly insulated), is lost in hot clinker (about 2%), or is lost up the stack. However, where stacks are on the inside building walls a portion of the stack heat is utilized below the roof level and does useful work.

[1]Here and subsequently, "cross section" of heaters means cross sectional area measured internally and horizontally at the firedoor.

TABLE 1.—COAL BURNING CAPACITY OF BOILERS AND FURNACES

Type of Heater	Burning Capacity, Lb. per Sq. Ft. of Horizontal Cross-Sectional Area per Hr., Up to
Cast iron sectional boiler	5
Cast iron round boiler, no interm. section	4
Cast iron round boiler, 1 intermediate section	5
Ditto, 2 intermediate sections	5½
Ditto, 3 intermediate sections	6
Steel heating boiler	10
Power boiler	20
Cast iron furnaces	4
Steel furnaces	8

Combustion Space

Coal underfed in stokers gives off its volatile element as coal gas as it reaches the zone of incandescence and this gas must have room to expand in order to burn efficiently. For coal having 30 to 40% volatile element, 1 cu. ft. of combustion space is needed for each 3 lb. per hr. of coal fuel. If less space is provided, part of the coal gas passes off unignited and is lost up the stack. If enough coal gas is lost unconsumed, the firebox temperature drops below ignition point of the coke, the combustion space fills, and all action stops. Often, due to lack of sufficient combustion space, where unburned coal gas constitutes 10% or more of the stack gases, it will ignite in the smokepipe, a bad fire hazard; unignited, it is a dangerous poison in concentrations as low as 2%.

All the firebox volume above the stoker tuyeres and ahead of the first gas passage is considered combustion space. Where a heater has adequate cross-sectional area for proper rate of coal feed but is short on combustion space, it is customary to pit the stoker, thereby providing whatever additional combustion space is needed.

The coal burning capacity of any heater when stoker fired is limited by its horizontal cross-sectional firebox area, as shown in Table 1.

The coal burning capacity of all heaters is further limited by the amount of combustion space which can be provided above the stoker tuyeres and ahead of the first gas passage. There must be at least 1 cu. ft. of combustion space for each 3 lb. per hr. of coal feed, this being the *rate of coal feed for any part of any hour* and not an averaged total per hour.

The Heating Load

Heating systems deliver the heat output of the heater to heating fixtures in spaces requiring heat. The summation of the maximum amount of heat which various spaces within the building require or which previously installed fixtures will deliver, plus losses which will occur between the heater and fixtures, is the heating load of the building.

In a well designed system the fixtures will be adequate to the load but a stoker survey is not complete unless both the load and fixtures are checked to assure that the system is well balanced to meet requirements.

Often stokers are expected to remedy defects in fixture capacity or balance, or to make an inadequate heater carry an impossible load and, unless the survey discloses existing defects, poor jobs will result. With the greatest care it is impossible always to forestall later trouble but careful surveying is the least part of the expense in a stoker business, the part most often slighted but which when properly done pays the largest return in proportion to money outlay.

The method of determining heater capacity has been discussed. Knowing what the heater can do, the next step is to estimate what is required of it—the load it must carry. The most convenient unit to use is the square foot of steam radiator as this has a definite B.t.u. value and is easily visualized. One square foot of steam radiator is 240 B.t.u. per hr. delivered by a fixture standing in 70° air. Instead of steam, the circulating medium may be hot water or warm air and, having determined the load in terms of required steam radiator, it is easy later to equate these mediums to steam heat equivalent or vice versa.

Any of three different methods can be used for estimating the building heat load. One is the 2-20-200, a crude method for getting a rough idea of the load; a short method which is more complicated than the 2-20-200 method, but is accurate within 10% of the actual load, and third, the detailed B.t.u. heat loss method.

The 2-20-200 Method.—For rough estimates this method is usable. It consists of totalling the area of outside windows and doors, and dividing by 2; totalling the area of outside walls and roofs and dividing by 20; and totalling the volume of the building to be heated and dividing by 200; then adding the three quotients. This sum is the number of square feet of steam radiator required for the building.

However, there are some exceptional conditions which should be taken into account in using this method. Twenty per cent should be added to the areas on the north wall, or if north is not the prevailing wind direction, to the walls facing the prevailing wind. Twenty-five per cent should be added for bathrooms, and 15% for rooms with open fireplaces.

Owner										Date				
Address										No. of Stories				
City										Estimator				
HEATED VOLUME - Cu. Ft.														
Story	Main Body Length x Width x Height = Volume				Ell Length x Width x Height = Volume				Ell Length x Width x Height = Volume				Total	
1														
2														
3														
Total Heated Volume														
WINDOWS and EXTERNAL DOORS*														
Side	Full Windows (Allow 1)		Half Windows (Allow ½)		Other Windows (Allow ¼, 2 etc.)		Doors (Allow 2)		Double French Doors (Allow 4)		Total No.			
North														
East														
South														
West														
* One equivalent window = 12 sq. ft.							Total Number of Equivalent Windows							
CALCULATION OF FACTOR							Expos. Factor ( ) x Total Factor ( ) = Overall Factor ( )							
	(Transmission Coeff.	x	Coeff. Multiplier)	= Factor			Heat Loss, B.t.u. per hr. per deg. (from fig. 2 and 3)							
Wall							Compactness (Deduct or add 5% see Explanation)							
Roof							Total (or Difference)							
Window Coeff. ( ) x No. Windows ( ) / Volume ( )							Add Unex. Floor Area ( ) x .2							
Total Factor							Total Heat Loss per degree							
Heat Loss B.t.u. per hr. per deg. temp. diff. ( ) x Temperature Difference ( ) = ( ) Total Heat Loss, B.t.u. per hr.														

Fig. 1. One type of condensed heat loss estimating sheet.

Short Form Method.—This is a condensed B.t.u. heat loss method and, as pointed out above, is accurate within 10% limits. Fig. 1 shows one form in common use for such estimates.[2]

Detailed B.t.u. Method.—This method consists of measuring the areas of outside walls, roofs, windows, doors, and floors, and multiplying these areas by the suitable coefficient depending on the type of wall, roof, or window, and by the difference in temperature existing under worst conditions between the inside and the outside, then totalling these products and dividing by 240 to obtain the square feet of steam radiator. This method has been so frequently described that it is unnecessary to give it further space here.[3]

[2] A complete description of this method appears on HEATING & VENTILATING'S Reference Data Sheets 33 and 34.

[3] Heat transmission coefficients for walls, ceilings, etc., appear on HEATING & VENTILATING'S Reference Data Sheets Nos. 71-78, inclusive. Design temperatures for different cities—that is, the difference in temperature between inside and outside—appear on HEATING & VENTILATING'S Reference Data 13A-14A and 57-58.

TABLE 2.—SQUARE FEET STEAM RADIATOR RATING PER SECTION, OLD STYLE COLUMNAR TYPE

No. of Columns	Overall Height in Inches									
	45	38	32	26	23	22	20	18	16	13
Single	—	3	2 1/2	2	1 2/3	—	1 1/2	—	—	—
Two	5	4	3 1/3	2 2/3	2 1/3	—	2	—	—	—
Three	6	5	4 1/2	3 3/4	—	3	—	2 1/4	—	—
Four	10	8	6 1/2	5	—	4	—	3	—	—
Hospital One	—	3	2 1/2	2	1 2/3	—	1 1/2	—	—	—
Hospital Two	5	4	3 1/3	2 2/3	2 1/3	—	2	—	—	—
Hospital Three	6	5	4 1/2	3 3/4	—	3	—	2 1/4	—	—
Hospital Four	10	8	6 1/2	5	—	4	—	3	—	—
Six col. window	—	—	—	—	—	—	5	4 1/2	3 3/4	3

TABLE 3.—SQUARE FEET STEAM RADIATOR RATING PER SECTION, TUBULAR TYPE

No. of Tubes	Overall Height in Inches								
	38	36	32	30	26	23	20	18	14
Three	—	3 1/2	3	2 3/4	2 1/3	2	1 3/4	—	—
Four	4 1/4	4	3 1/2	3 1/4	2 3/4	2 1/2	2 1/4	—	—
Five	5	4 3/4	4 1/3	4	3 1/2	3	2 2/3	2 1/2	—
Six	6	5 3/4	5	4 3/4	4	3 1/2	3	—	—
Seven	7 1/2	6 3/4	6 1/4	5 1/2	4 3/4	4 1/4	3 2/3	3 1/2	2 1/2

TABLE 4.—STEAM RATINGS OF WALL RADIATORS

Dimensions, Inches	Sq. Ft.
13 5/16 x 16 5/8	5
13 5/16 x 21 7/8	7
13 5/16 x 29 1/16	9

Having calculated the amount of steam radiator surface necessary for the building it remains only to check whether the installed surface is equal to that calculated. Where a hot water system is used instead of steam, the amount of radiator surface installed will be 60% more for an open system, and 30% more for a forced water system than the steam surface. If a gravity warm air system is used, about 1¾ sq. in. of duct should be provided for each square foot of steam radiator calculated, provided the system is a gravity one. Vapor systems can be figured exactly the same as steam.

In addition to these points there are a number of special cases which should be noted and on which information should be obtained. These include use of so-called indirect radiators and radiators which are enclosed. The newer type convectors, which are a modern type of enclosed radiator but which give off the greater part of their heat by convection rather than by radiation, should be checked and the rating determined from the manufacturer. Tables 2, 3, 4, and 5 give information on the square feet of steam radiator equivalent of old type columnar radiators, the newer type tubular radiator, wall radiator, and pipe coils. For any radiator which does not fall in these classifications, obtain information on the rating from the manufacturer.

Other points where special conditions apply are rooms which are maintained at temperatures other than 70°, lobbies and stores where doors are opened frequently, and rooms with extremely high ceilings. If either the short form or detailed heat loss methods are used for calculating the heat requirements, tables of coefficients mentioned vary according to whether the building is insulated and whether the windows are weatherstripped, etc. If the approximate 2-20-200 method is used, however, allowance must be made for such special conditions.

Pipes leading from the heater to the radiators should be examined. Where steam lines in cellars are not too long, are well insulated, and have no traps or pockets for accumulating water from condensation, line losses will not be excessive. Warm air ducts for gravity systems should have an upslope all the way of at least ½ in. per ft. of run. Ducts for forced circulation warm air are usually level.

Where heaters in an old or remodelled building are to be stokered the survey should be supplemented by consultation with the heating contractor or with a heating man who has done work on the system. Also, in steam plants where new piping has been installed, it is important to obtain competent assurance that all oil has been thoroughly eliminated, as oil in the boiler water will prevent proper steaming. The quality of the radiator valves for air removal is an important consideration. In one-pipe steam plants, pounding and irregular heating will occur unless water from condensation can run back freely toward the steam inlet.

Service Hot Water

In most residences and apartment houses, where service hot water is heated by an indirect heater attached to the boiler, the additional load may be taken as roughly 20% of the building heating load. In schools, garages, public buildings, office buildings, and factories it is necessary to estimate the service hot water load on the basis of maximum requirement in gallons per hour. The accurate determination of the hot water load is, like the calculation of heat loss, fairly complicated.[4]

Service hot water load is often expressed as equivalent steam radiation. Thus, where a stoker fired heater is 60% efficient, 6000 B.t.u. of a 10,000 B.t.u. per lb. coal will be available, and each pound burned per hour will supply 6000 ÷ 240, or 25 sq. ft. of *total* steam radiator or, with 20% line loss, 20 sq. ft. of *installed* radiator. With submerged hot water heating coils, 1 lb. of coal will heat 9 gal. of service hot water. Hence, roughly, 1 gal. per hr. of service hot water is the equivalent of 2 sq. ft. of steam radiator. By a similar calculation, with the coil in the firebox, 1 gal. per hr. is equivalent to a load of 3 sq. ft. of installed steam radiator.

Selection of Stoker

Knowing the heating load which must be provided for, it is an easy matter to select the proper size stoker. The total load consists of the radiators (or duct area), the losses which may be expected between the heater, and the service hot water, if any. The sum of these, expressed as square feet of steam radiator, may be directly equated to pounds of coal per hour. A square foot of steam radiator is equal to 240 B.t.u. per hr., and where 10,000 B.t.u. per lb. screenings will be the stoker fuel and the heater efficiency 60%, each pound per hour of coal feed will handle 6000 ÷ 240, or 25 sq. ft. of total radiator. If the heater will properly handle the coal at the required rate and if the lines from the heater to the fixtures and the fixtures themselves are adequate, good results may be expected. If any part of the system, heater, lines, or fixtures is inadequate and the deficiency cannot be remedied, a stoker installation is not justified, as there is a reasonable expectation that service will eat up profits and, still more important, sales will be lost elsewhere due to unfavorable reports from the job in question.

If, as sometimes happens, a steam boiler to be stokered is much too large for the load, it is advisable to put in a stoker adjusted to the heater rather than to the load. The reason is that before steam can be generated, not only must the boiler water be heated to 212° but a larger amount of heat must be added to the water to cause it to change its state from liquid to steam. If the body of water in the boiler is too great,

[4]For further information on this subject the reader is referred to Heating & Ventilating's Reference Data Sheets 41-46, inclusive.

it is difficult for the stoker to get the water steaming although, once pressure develops, it is easy to hold steam.

It is a temptation to meet a competitor's price by skimping on the stoker size, but it is poor business in the end. If it becomes certain that the order will go to a competitor, it is best to do all possible to assure that the competitive stoker goes in large enough, as a satisfactory job, even that of a competitor, helps build up stoker business for all, while every poor job discourages sales for all makes of stokers.

Controls

There are two controls which should go into every job—the hold fire and limiting controls. Additional for large jobs is usually a day-and-night starting and stopping clock and, for small jobs, a room thermostat. There should also be an alarm arrangement to show stoppage of coal feed. The hold fire should be an adjustable interval operating clock cut into the primary electric circuit and so wired that it will not function if the limiting control on the heater has the stoker shut off. Where a room thermostat is used it should be connected in series with the limiting control so that both must be calling to start the stoker.

The no-coal-feed alarm may work when the fire burns low in the retort but, for simplicity, the best is a motor overload cutout switch which will signal when it cuts out. The fan load on the motor is about five times the coal feed load and is highest when the fan is delivering maximum volume. When back pressure against the fan drops too low, the fan delivery volume builds up the motor load (unless there is a functioning automatic damper in the air duct) and, when the coal feed screw becomes blocked, will increase the motor load tremendously until the safety device releases. A high grade overload switch which will cut out and sound an alarm is recommended.

TABLE 5.—MULTIPLYING FACTORS FOR CONVERTING WALL COILS TO SQ. FT. OF RADIATOR
(Table applies to horizontal pipes in vertical coils)

To use, multiply the length of the coil (not the length of the pipes) by these factors. The product is the square feet of steam radiator.

No. of Pipes in Coil	Pipe Diameter, Inches		
	1	1¼	1½
One	0.73	0.90	1.02
Two	1.40	1.72	1.92
Four	2.43	3.01	3.39
Six	3.13	3.87	4.38
Eight	3.59	4.44	5.00
Ten	4.03	5.00	5.62
Twelve	4.48	5.55	6.25

For ceiling coils (horizontal pipes in horizontal coils), use multiplying factors of 0.70, 0.85, and 0.96 for 1, 1¼, and 1½-in. pipes, respectively, for single pipe 1 ft. long. Multiple pipes in proportion.

Stack Conditions

A complete picture at the point where gases leave the heater, and showing the temperature, composition of gases and draft, is the best possible check on stoker operation. Overheated stacks mean a useless excess coal feed or too much stack draft; the presence of appreciable carbon monoxide indicates insufficient combustion space or insufficient air for proper combustion; smoke indicates either insufficient air or insufficient combustion space, while an entirely clear stack means excess air; low carbon dioxide and excess oxygen show either too much air from the fan or air leakage above the fire.

For heaters, stoker fired, draft below the damper should not exceed 0.2 to 0.4 in., carbon monoxide should be under 1%, oxygen should not exceed 7%, carbon dioxide should be better than 7%, stack temperature should be below 550°, and gases leaving the stack should have a hazy appearance. A low priced, 3-cell, portable Orsat, a stack thermometer and draft gage should be part of the service equipment of any stoker dealer who expects to do considerable business. This equipment is very simple to use and will soon repay its cost by limiting repeat service calls.

Unit Heater Dries Packaged Fuel

WHEN the Avers Fuel Corp., Brooklyn, N. Y., started to manufacture Pocahontas coal "bricks," it experienced difficulty in eliminating from the finished packages the water used as part of the binder in the briquetting process. It proved necessary to erect a large dry room and install a unit heater at one end for drying and an electric exhaust fan at the opposite end of the drier to remove the moisture laden air.

Following the forming of the Pocahontas bricks, each weighing 15 lb., by hydraulic pressure, the bricks are wrapped in heavy brown paper and appear to be ready for sale. It is, however, necessary to dry the fuel to obtain good combustion as a ton of packaged coal holds, after forming in the press, about 60 lb. of water. Several ineffectual drying methods were tried. Finally, a wooden cabinet some 20 ft. long by 12 ft. wide by 6 ft. high was built. Four trucks, each holding about 1 ton of coal packages, are rolled into the drier and the steam turned into the unit heater, the heater's fan blowing heat into the dry room to maintain 140° as a maximum drying temperature.

The warm air is introduced at one end near the top of the drier. At the opposite end, near the floor, a 16-in. propeller fan extracts the saturated air, discharging it outdoors. Between four and five hours' operation are necessary to provide adequate drying. The management states that when coal bricks now leave the drier they are practically bone dry.

Chicago Ventilating Contractors Claim Air Conditioning Work

IN recent years a number of movements have either been originated by or begun in Chicago. That city was the first to have local air conditioning standards; a decade ago the boiler rating sheets of the Chicago Master Steamfitters' Association were so widely used that the idea was adopted nationally; more recently Chicago was selected as the site of the opening gun of the radiator heat campaign.

Last month from that city came a Declaration of Principles endorsed by the Ventilating & Air Conditioning Contractors Association of Chicago[1] in which this group stated its policy and pledged itself to trade protection (as well as professional protection); called for separate specifications on ventilating and air conditioning; recognized special practices in bid taking and awarding, and claimed certain items of equipment and certain work as definitely belonging to the ventilating and air conditioning contractors. In addition, it lists those to whom the member contractors will submit proposals for installations.

Following are extracts from the Association's Declaration of Principles which include about 90% of the material contained in the formal declaration.

Article I

We declare that the science of mechanical ventilation and air conditioning is technical, intricate, and scientific, requiring a high degree of training, experience, and practical knowledge of the fabrication, erection and installation of such systems, together with engineering and state and city code requirements as well; and we hold therefore that it is the first duty of every ventilating and air conditioning contractor to study the technique of his industry, to know his business thoroughly and not superficially, because only through knowledge of facts can a business man operate without injury to himself and to his trade. Membership in this association is and shall be predicated on these requirements.

Article II

We further declare that the members of this Association believe in trade protection. By trade protection is meant that manufacturers of and wholesale dealers in materials and apparatus which enter into the construction of ventilating and air conditioning systems and other work done by those engaged in this craft shall sell such material and apparatus for use in such work to those only who are regularly established in this line of business and shall leave the preparation of plans and specifications to qualified architects or engineers. We further declare that fair trade practice should not permit the invasion of the building construction industry as contractors by manufacturers, their agents or brokers. We shall endeavor by every legal and legitimate means to further trade protection.

Architects and engineers have for a long time recognized the merits of the aforemade declarations and have practiced a proper observance of their responsibilities to their clients; first, by writing separate and distinct specifications for all ventilation and air conditioning work; second, by letting all contracts for such work individually and separate from all mechanical or other contracts.

Co-operation in this matter accomplishes the following results: First, it discourages an unfair practice of letting ventilation and air conditioning work to intermediate contractors and others whose relationship to it, if any, is merely one of agency or brokerage; second, they have effected a material saving to their clients of unfair overhead charges and profits taken ordinarily by some of such agents or brokers who render no practical service in this work but rather subject this most important element in modern building construction to a destructive process of specification muddling, causing the use of lighter gauges of sheet iron inconsistent with underwriters and State and City Code requirements, omitting safety factor margins and allowances in fans, blowers, motors and other equipment often found necessary for increased needs or alterations, and other kindred abuses, often substituting a substandard or inferior installation without assuming corresponding responsibilities caused thereby. In most cases the original cost to the owner or prices paid to such brokers would, in direct letting of such contract work, be ample for high class systems, as originally specified by qualified architects and engineers. . . .

Article III

We further declare that the estimating and compilation of a bid on ventilation and air conditioning work is laborious and entails considerable cost to the bidding ventilating and air conditioning contractor. We recognize two forms of taking bids as ethical and fair; first, public opening of bids and letting of contracts in accordance with the findings to the lowest acceptable bidder; second, taking private competitive bids from a suitable number of original acceptable bidders, all bids being held strictly confidential and awards being made to the lowest acceptable or responsible bidder on the basis of the original bid without meeting lower bids of non-acceptable bidders. Fair practice should not permit the changing or altering of the original bid, cutting such a bid, or receiving such a changed bid,

(Continued on page 74)

[1]Whose membership consists of 24 firms. The Association was established in 1924. Its officers are: President, P. R. West, Western Ventilating and Engineering Co.; vice-president, E. B. Brown, Reynolds Corp.; treasurer, Otto Wendt, Wendt and Crone; secretary, Julius Vogt, Standard Sheet Metal Works; executive secretary, L. W. Rogers.

New Value of Coefficients for

Photo courtesy Portland Cement Assn.

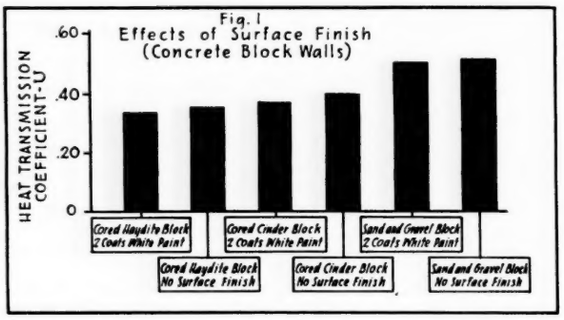


Fig. 1
Effects of Surface Finish
(Concrete Block Walls)
HEAT TRANSMISSION COEFFICIENT-U
.60
.40
.20
0
Cored Haydite Block 2 Coats White Paint
Cored Cinder Block 2 Coats White Paint
Sand and Gravel Block 2 Coats White Paint
Cored Haydite Block No Surface Finish
Cored Cinder Block No Surface Finish
Sand and Gravel Block No Surface Finish

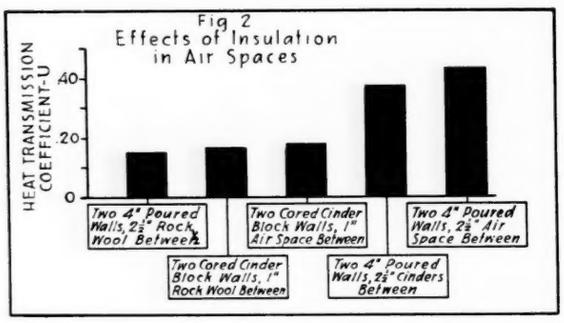
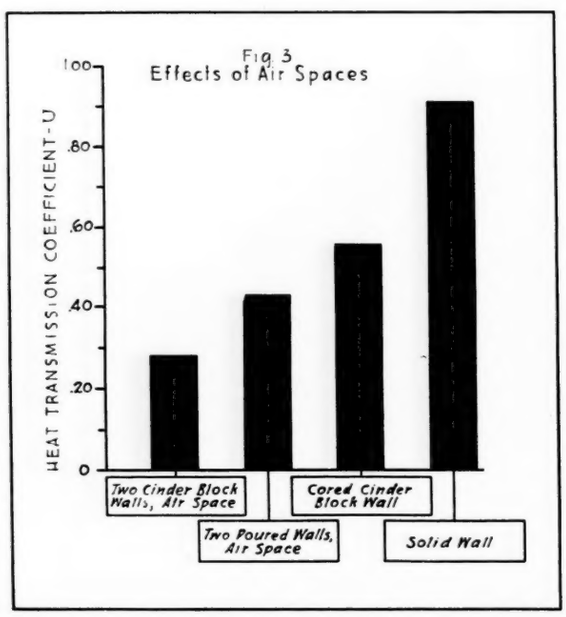


Fig. 2
Effects of Insulation in Air Spaces
HEAT TRANSMISSION COEFFICIENT-U
.40
.20
0
Two 4" Poured Walls, 2½" Rock Wool Between
Two Cored Cinder Block Walls, 1" Rock Wool Between
Two Cored Cinder Block Walls, 1" Air Space Between
Two 4" Poured Walls, 2½" Cinders Between
Two 4" Poured Walls, 2½" Air Space Between



CONCRETE is a material with high structural strength and other good qualities, but its low heat resistance makes it necessary to give special consideration to the methods of using such walls in buildings where heat transmission is of any importance. Various methods can and have been used to improve this property of concrete walls, and a recent study was concerned with investigating some of these.

The heat transmission of a concrete wall depends on the type of aggregate used, surface finish, air space construction, and amount and type of insulating material used.

For convenience in interpreting the results presented in the paper reporting this study, this article has been prepared and divided into two parts. The first part, illustrated by Figs. 1—6, shows the effects of changes in the variables which affect the heat transmission of concrete walls so that these effects can be noted almost at a glance. The second part is summarized in Fig. 7, a working tabulation which contains the data from the tests arranged in convenient and usable form. It will be noted that Fig. 7 is arranged, for the sake of consistency, in the same form as the heat transmission tables for other types of walls and roofs which have recently appeared on these pages.

Effects of Surface Finish

See Fig. 1. The effect of surface finish on a wall may be to reduce the normal air filtration into and out of the wall and thus improve the internal conductivity by eliminating convection currents, or to change the character of the surface and thereby change the surface coefficient of conductance. The experiments showed that the conductance (C) of a concrete wall was decreased by an amount between 3½% and 9½% by painting the natural cement surface with two coats of waterproofed cement paint. However, this resulted in a reduction in the overall heat transmission coefficient (U) only of an amount between 2% and 6½%.

Effects of Aggregates

See Fig. 4. In general, low density aggregates[1] will result in lower thermal conductivity than will high density aggregates. This, however, is not always the case as low density may be accompanied by an excessive porosity which will allow the heat to be transferred through the wall by air circulation. In this investiga-

[1]The term "aggregates" is here used to denote the materials other than cement and water making up the concrete. Haydite, one of the aggregates tested, is a lightweight cellular, inert, burned-shale material.

Heat Transmission Concrete Walls

This article is based on data and conclusions presented in a paper resulting from research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the University of Minnesota and the Portland Cement Association. Authors of the paper were Professors F. B. Rowley, A. B. Algren, and Clifford Carlson, all of the University of Minnesota.

tion sand, crushed limestone, sand and gravel, cinders and Haydite were used as aggregates.

Effects of Air Spaces

See Fig. 3. Air space constructions may be used for insulating purposes. In the case of poured concrete walls, the air space can be that existing between two separate parallel walls or, in the case of concrete blocks, the air space can consist of open cores inside the block. In some cases concrete walls with cores were used in double form with an additional air space between the two walls.

Effects of Insulation in Air Spaces

See Fig. 2. This drawing shows the comparison of different combinations of walls with air spaces, but having no insulation, while Fig. 5 shows the effects of putting different kinds of insulation in the cores of concrete blocks. Fig. 6 shows the effects of placing insulation between the ribs of a poured 2-in. wall having 4 x 4 in. ribs.

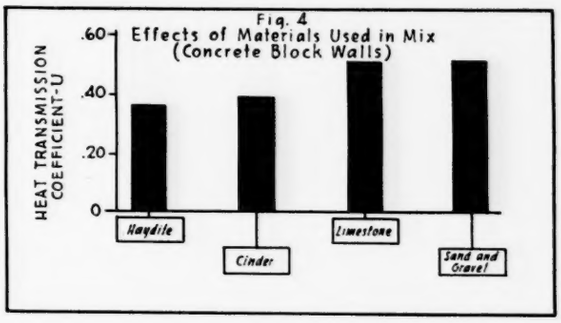
Summary

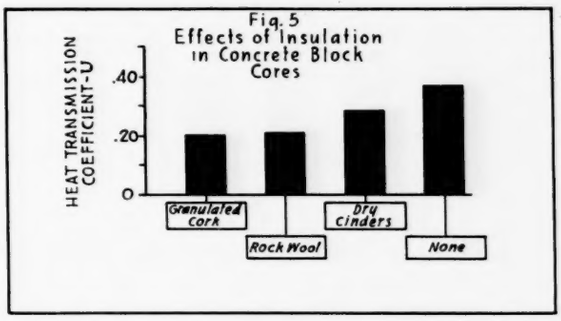
The tests showed definitely that the thermal conductivity of poured concrete walls is directly proportional to the density and air weight per cubic foot of the wall. They also showed that for concrete with average aggregates (but not considering those with special low density aggregates) a conductivity value (k) of 12.0 to 13.0 is reasonable.

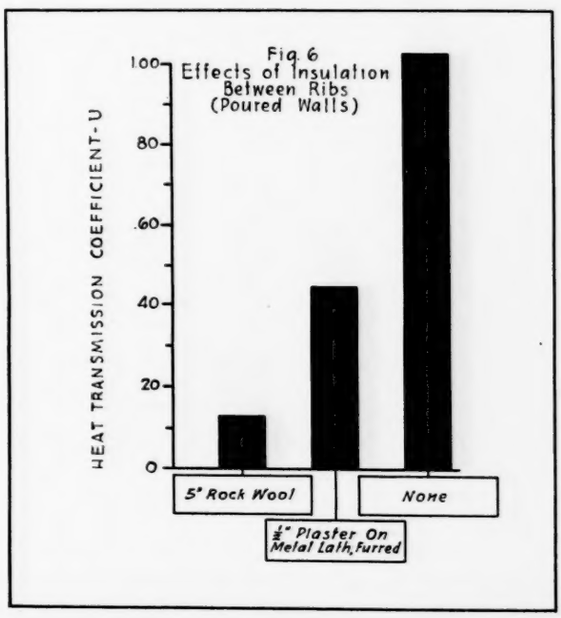
Lightweight aggregates are more effective in reducing the heat transmission of poured walls than they are in reducing the coefficient of concrete block walls with cored-out sections. The reduction in conductance of a masonry wall built with 12-in. blocks over that of a similar wall built with 8-in. blocks depends upon the density of the aggregate, but for average construction it is less than 15%, whereas for a monolithic wall the reduction would be 50%. This difference is due largely to the shape and location of the air spaces in the design of the blocks.

In concrete masonry or monolithic walls built up with parallel slabs for the purpose of obtaining an air space between or for the use of insulating material between the two slabs, special consideration must be given to the conductivity of any tie rod or support between the slab.

The effectiveness of an insulating material when placed in the cores of concrete blocks will depend upon the thermal conductivity of insulation, thermal con-







CONCRETE WALLS

Figures are in B.t.u. per hour per square foot per degree temperature difference, air to air, still air inside, 15 m.p.h. wind outside.

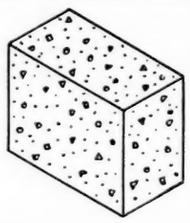
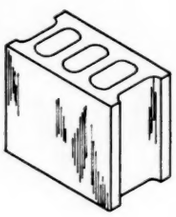
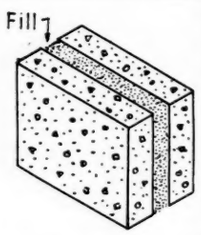
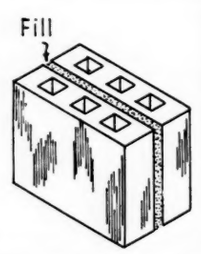
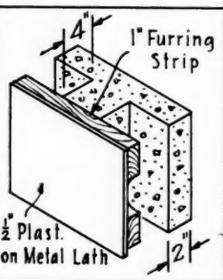
Construction No.	Wall Type	Aggregates Composed of	Total Wall Thickness, In.	Exterior Finish	Interior Finish	Insulating Fill: None	Insulating Fill: Cinders	Insulating Fill: Cork	Insulating Fill: Rock Wool	Thickness of Fill Space, In.
131	Poured Concrete 	Sand and Limestone	4	Natural	Natural	0.87	–	–	–	–
132		Sand and Gravel	4			0.89	–	–	–	–
133		Cinder	4			0.68	–	–	–	–
134		Haydite	4			0.54	–	–	–	–
135		Sand and Cinder	4			0.80	–	–	–	–
136	Concrete Blocks 	Cinder	8	Natural	Natural	0.40	–	–	–	–
137					½" Plaster Appl. Direct	–	–	–	0.21	4¾
138				White Paint	Natural	0.37	0.28	0.20	0.21	4¾
139					½" Plast. on Metal Lath Furr. 1"	–	–	–	0.17	4¾
140				½" Plast. Appl. Direct	½" Ins. Bd. Furred 1"	–	–	–	0.15	4¾
141			12	Natural	Natural	0.37	–	–	–	–
142		Haydite	8	Natural	Natural	0.36	–	–	–	–
143				White Paint		0.33	–	–	–	–
144		Sand and Gravel	8	Natural	Natural	0.52	–	–	–	–
145				White Paint		0.51	–	–	–	–
146			12	Natural	Natural	0.48	–	–	–	–
147		Cinder Part. Tile	4	Natural	Natural	0.56	–	–	–	–
148	Double 4" Poured Conc. 	Sand and Gravel	10½	Natural	Natural	0.43	0.38	–	0.15	2½
149	Double 4" Conc. Blocks 	Cinder Partition Tile	9	Natural	Natural	0.28	–	–	0.18	1
150	2" Poured, 4"×4" Ribs 	Sand and Gravel	7½	Natural	½" Plaster on Metal Lath Furred 1" on Ribs	0.45	–	–	0.13	5

Fig. 7. Heat transmission coefficients for various types of concrete walls. Note that the values given are from actual tests and not calculated.

ductivity of the aggregate, and design of the cored spaces in the block. Consequently, it follows that in order to get the full value of an insulating material when placed in any wall, it must be so placed as to effectively block off all paths of heat flow which may join the two surfaces of the wall.

GETTING DOWN TO FUNDAMENTALS

11—Estimating Sun Heat Through Walls and Roofs

PREVIOUS articles in this series have dealt with the data and calculating methods available and used in attempting to prepare estimates of heat quantities which are encountered in the ordinary job. In addition to these quantities there are situations where estimates are virtually meaningless unless the quantities due to other exchanges of heat are included.

We have already considered Filtration Heat, Transmitted Heat, and Heat from Ventilation Air. Among those quantities which also sometimes affect the estimate are sun heat, electrical and mechanical heat, heat due to humidity changes, and various others. Now we can turn attention to some of these quantities, first giving attention to the subject of Sun Heat.

In considering the subject the same general method used in previous articles will be followed. This consists of a short statement of the elementary theory, then a collection of what appear to be the best available data, and clarifying comments on application or use of the data. In the case of sun heat, methods of applying the available data are by no means uniform, and there are still many opinions as to how estimates of this quantity should be made.

When Sun Heat is Important

It is well to keep in mind that sun heat is usually considered as of importance only in estimating the summer cooling load. While there is evidence to show that this heat may affect the winter peak load, and probably does in some localities, still it is practically universally disregarded in winter peak estimates because it is an uncertain quantity and because it is likely to decrease the peak rather than increase it. In summer, though, sun heat may become so important a fraction of the total peak that some allowance for it simply must be included in many cases.

Sun heat reaches a building and adds to the cooling load (a) by way of the solid walls and roof exposed to the sun, and also (b) by way of glass surfaces usually in the form of windows or skylights on which the sun shines. As glass permits a considerable amount of the sun energy to pass through as radiant heat while solid walls and roofs do not, it is necessary to use different methods and different data in estimating the sun heat for the one case than for the other. This distinction naturally divides the subjects into two separate parts: (1) estimating sun heat through walls and roofs, and (2) estimating sun heat through glass windows and skylights. We will now consider only the first of these, sun heat through walls and roofs, leaving the second to be discussed later.

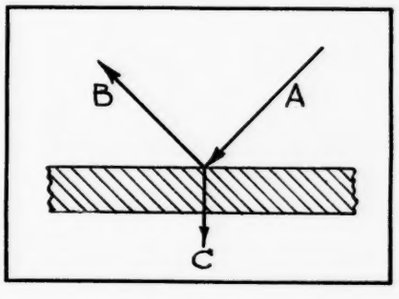


Fig. 1.

How Sun Heat Reaches the Interior

A statement of the general theory concerning the way sun heat reaches the building interior through walls and roofs is helpful. Fig. 1 shows the usually accepted explanation. Here radiation from the sun strikes the surface of the solid building wall or roof and part of it is immediately reflected along *B*. Some, however, is absorbed by the surface and the temperature of the surface is increased. This increase in surface temperature sets up a path of temperature difference and heat starts flowing toward the cooler surface inside. Eventually some of this heat appears at the inside surface, is conducted off the inside surface by the air inside. This heat is that which goes to add to the cooling load and is the quantity the amount of which it is desired to estimate.

From Fig. 1 several points are at once apparent. One is that the sun heat is always less than the amount of radiant heat given off by the sun, for only a part of this total is absorbed by the solid. A second is that for a given amount of radiant heat from the sun the sun heat with which we are concerned will depend on the angle at which the radiant heat strikes the surface. A third is that the sun heat is really a form of transmitted heat so far as walls and roofs are concerned.

Knowing these things how can we go about actually estimating the amount of the sun heat?

Two Methods of Estimating

Two possible methods suggest themselves at once. One is to use the usual formula for transmitted heat. The other is to learn the amount of radiant heat emitted by the sun and to decrease this by suitable multiplying factors since the sun heat is always less than the radiant heat of the sun.

In order to use the first method it is necessary to know the transmission coefficient for the construction, the area, and the surface temperature of the two surfaces of the construction affected by the sun. Absence of information principally concerning the outside surface temperature attained under the influence of heat from the sun makes use of this method difficult at present.

The second method makes use of the following formula

$$H_s = A\ I\ b\ g \qquad (4)$$

where:

H_s = Sun heat, B.t.u. per hr.
A = Area affected by sun, sq. ft.

I = Intensity of solar radiation striking the area, B.t.u. per hr. per sq. ft.
b = Decimal part of I absorbed by the surface.
g = Decimal part of b transmitted to the inside.

This formula becomes usable as soon as values of the various factors can be fixed, and we can now proceed to obtain suitable values.

Finding Values for Solar Intensity

The intensity of the solar radiation striking the surface, the quantity I in the formula, is troublesome to evaluate. The principal difficulty is due to the fact that an adequate amount of study has not been devoted to it. Moreover, it is an extremely variable quantity since it depends on the latitude, the season of the year, the hour of the day, the condition of the atmosphere, and on the position and orientation of the surface relative to the sun.

While a direct measurement of the amount of solar radiation striking a horizontal surface or a surface normal to the sun is not particularly difficult with existing apparatus, such measurements have not been generally taken. There are a few weather stations equipped with suitable apparatus and the data from these stations provide nearly all of the scanty information available. It appears that the maximum amount of solar radiation is received at or about noon sun time; that it builds up rapidly after sunrise and falls off after noon in much the same way that it increases during the morning hours. Atmospheric conditions, particularly the humidity in the air and the amount of dust and smoke present greatly affect the intensity of the radiation received at a particular time and place. The net result of these complications is that the existing field data are full of confused and confusing details and attempts to apply any rigid rules to them are discouraging and of doubtful value.

It appears, therefore, that the best that can be done at present is to set up values of the intensity of solar radiation received on a surface normal to the sun which, while not correct, can at least be used temporarily until better information is at hand.

Since what is usually wanted in preparing an estimate is some reasonable allowance for sun heat at the time of the summer peak load, we can dismiss for the time being any consideration except at that time. This simplifies the problem and makes it necessary to consider only the solar radiation for clear days in late summer when the seasonal peak on the cooling plant is likely to take place. What information there is indicates that in the north latitude range of 25° to 45° within which virtually all of our cities of large population lie, the intensity of solar radiation received on a surface normal to the sun on clear days in late summer can be reasonably represented by Table A.

TABLE A

Hour	B.t.u. Per Sq. Ft	Hour	B.t.u. Per Sq. Ft.
6 a.m.	120	1 p.m.	290
7 "	200	2 "	280
8 "	250	3 "	260
9 "	270	4 "	240
10 "	290	5 "	200
11 "	295	6 "	120
Noon	300		

The values in this table, however, are not those for practical use in estimating for they show the solar radiation received on a surface normal to the sun and not on vertical walls or flat roofs, which surfaces will present various orientations to the sun position. These values can be made the basis for a working tabulation of practical values provided they can be modified to take into account the latitude and the angular position of the surface on which the solar radiation is to be estimated.

Several years ago Walker and Hendrickson[1] prepared tables giving factors by which hourly values of the solar radiation received on a surface normal to the sun in various latitudes and at various seasons of the year can be multiplied to determine the intensity of solar radiation striking horizontal or vertical surfaces at various compass exposures to the position of the sun.

Table 1 has been prepared by combining these factors for the summer period with the hourly values of solar radiation given in Table A for four different north latitudes within the range of which nearly all of our largest cities lie. It should be noted carefully that the values given in the table are not strictly accurate but it is believed that they are sufficiently accurate to meet many practical estimating needs. At any rate they illustrate hourly variations in intensity and make it apparent which walls are the important ones to consider in preparing estimates. They also bring out well the point that flat roofs exposed directly to the sun receive appreciably more solar radiation per unit area exposed than do vertical side walls. Sloped roofs receive amounts of solar radiation not listed in the table but in some cases falling intermediate between the amount received on a flat roof and that received on a vertical wall. Amounts received on sloped roofs may be approximated from the figures given in the tables but any such estimating should be done with care. Table 1 can thus be considered as showing the best available data for establishing values of I in equation (4).

Absorption of Solar Energy

It has been noted before that only a part of the solar radiation reaching a wall or roof structure is absorbed by it, for a considerable part is reflected at once and never gets a chance to penetrate the structure and add to the cooling load. The factor b in equation (4) allows for this and is sometimes called the absorption coefficient. Experiments have shown that the value of this coefficient for the materials used in building walls and roofs or in covering or coating them varies from about 0.20 to 0.90. In other words, from 20% to 90% of the solar radiation I in Table 1 is converted into sensible heat as it strikes the surface. The rest is reflected back into the atmosphere at once.

Many investigators have reported values of the absorption coefficient but the studies have not always been made to determine the values as applied to solar energy. Table 2 lists values collected from several sources but principally from studies of the Building Research Board of Great Britain. It is believed that

[1]See Heating & Ventilating, November, 1932, pages 14-21.

TABLE 1.—INTENSITY (I) OF SOLAR RADIATION IN SUMMER STRIKING WALLS AND ROOFS

(Figures are in B.t.u. per hour)

	Time of Day	SIDE WALLS (VERTICAL) Facing compass point below								FLAT ROOFS (Horizontal Surface)
		E	SE	S	SW	W	NW	N	NE	
30° North Latitude	6AM	113	55					35	105	20
	7	180	104					35	153	75
	8	204	133					15	155	145
	9	180	134						120	200
	10	136	120	32					73	254
	11	72	85	47					18	283
	Noon		37	52	37					296
	1			46	82	70	18			277
	2			31	115	132	71			245
	3			10	130	173	116			195
	4				128	195	148	15		138
	5				105	182	153	35		75
	6				55	113	105	35		20

Austin, Tex. Baton Rouge, La. El Paso, Tex. Galveston, Tex. Houston, Tex. Jacksonville, Fla. Laredo, Tex. Mobile, Ala. Montgomery, Ala. New Orleans, La. Pensacola, Fla. San Antonio, Tex. Savannah, Ga. St. Augustine, Fla. Tampa, Fla.

Cities listed above lie between 27° 30′ and 32° 30′ North Latitude

35° North Latitude	6AM	113	50					33	103	24
	7	180	102					28	148	79
	8	204	138	8				10	146	145
	9	180	147	27					108	200
	10	136	140	54					58	250
	11	72	112	71	10				12	277
	Noon		71	77	55					290
	1		11	70	99	70	12			277
	2			52	129	132	56			242
	3			26	140	173	104			192
	4			7	136	195	140	10		138
	5				109	181	148	28		60
	6				56	113	103	33		24

Atlanta, Ga. Birmingham, Ala. Charleston, S. C. Chattanooga, Tenn. Dallas, Tex. Fort Worth, Tex. Knoxville, Tenn. Little Rock, Ark. Long Beach, Calif. Los Angeles, Calif. Memphis, Tenn. Nashville, Tenn. Norfolk, Va. Oklahoma City, Okla. Phoenix, Ariz. Raleigh, N. C. San Diego, Calif. Tulsa, Okla.

Cities listed above lie between 32° 30′ and 37° 30′ North Latitude

40° North Latitude	6AM	113	45					31	102	26
	7	180	100					21	143	81
	8	204	143	15				5	137	145
	9	180	160	44					96	197
	10	136	162	75					43	244
	11	72	141	94	22				6	270
	Noon		105	102	72					282
	1		21	92	115	70	6			265
	2			73	144	134	42			236
	3			43	152	172	92			190
	4			14	145	195	132	5		138
	5				114	180	143	21		81
	6				58	113	102	31		26

Akron, Ohio. Baltimore, Md. Berkeley, Calif. Boston, Mass. Camden, N. J. Carson City, Nev. Charleston, W. Va. Cheyenne, Wyo. Chicago, Ill. Cincinnati, Ohio. Cleveland, Ohio. Columbus, Ohio. Dayton, Ohio. Denver, Colo. Des Moines, Iowa. Detroit, Mich. Evansville, Ind. Fort Wayne, Ind. Harrisburg, Pa. Hartford, Conn. Indianapolis, Ind. Kansas City, Mo. Louisville, Ky. Newark, N. J. New York, N. Y. Oakland, Calif. Omaha, Neb. Paterson, N. J. Peoria, Ill. Philadelphia, Pa. Pittsburgh, Pa. Providence, R. I. Reading, Pa. Richmond, Va. Sacramento, Calif. Salt Lake City, Utah. San Francisco, Calif. Scranton, Pa. South Bend, Ind. Springfield, Ill. Springfield, Mass. St. Louis, Mo. Toledo, Ohio. Topeka, Kan. Washington, D. C. Wichita, Kan. Youngstown, Ohio.

Cities listed above lie between 32° 30′ and 42° 30′ North Latitude

45° North Latitude	6AM	113	42					29	100	30
	7	180	99					14	138	83
	8	204	148	22					128	143
	9	180	173	62					83	192
	10	136	183	97					28	247
	11	72	167	118	33					260
	Noon		136	130	90					270
	1		32	116	132	70				257
	2			93	158	132	27			229
	3			59	164	172	80			185
	4			22	154	195	123			138
	5				118	182	138	14		83
	6				59	113	100	29		29

Albany, N. Y. Astoria, Oreg. Augusta, Me. Bismarck, N. Dak. Buffalo, N. Y. Duluth, Minn. Flint, Mich. Grand Rapids, Mich. Helena, Mont. Madison, Wis. Milwaukee, Wis. Minneapolis, Minn. Montreal, Canada. Ottawa, Canada. Portland, Me. Portland, Oreg. Rochester, N. Y. Seattle, Wash. Spokane, Wash. St. Paul, Minn. Syracuse, N. Y. Tacoma, Wash. Toronto, Canada. Utica, N. Y.

Cities listed above lie between 42° 30′ and 47° 30′ North Latitude

TABLE 2.—SOLAR RADIATION ABSORBED BY VARIOUS SUBSTANCES.

Brick	
Glazed, white	0.26
Glazed, ivory to cream	0.35
Common, light red	0.55
Common, red	0.68
Wire-cut, red	0.52
Mottled purple	0.77
Blue	0.89
Limestone	
Light	0.35
Dark	0.50
Sandstone	
Light fawn	0.54
Light grey	0.62
Red	0.73
Marble	
White	0.44
Dark	
Granite	
Reddish	0.55
Metals	
Steel, vitreous enameled, white	0.45
Steel, vitreous enameled, green	0.76
Steel, vitreous enameled, dark red	0.81
Steel, vitreous enameled, blue	0.80
Galvanized iron, new	0.64
Very dirty	0.92
White, washed	0.22
Copper, polished	0.18
Tarnished	0.64
Lead sheeting, old	0.79
Paints	
Aluminum	0.54
Cellulose, white	0.18
Cellulose, yellow	0.33
Cellulose, orange	0.41
Cellulose, signal red	0.44
Cellulose, dark red	0.57
Cellulose, brown	0.79
Cellulose, bright green	0.79
Cellulose, dark green	0.88
Cellulose, dark blue	0.91
Cellulose, black	0.94

Roofing Materials	
Asbestos-cement, white	0.42
Asbestos-cement, 6 months' exposure	0.61
Asbestos-cement, 12 months' exposure	0.71
Asbestos-cement, 6 years' exposure, very dirty	0.83
Asbestos-cement, red	0.69
Asphalt, new	0.91
Asphalt, weathered	0.82
Bitumen-covered roofing sheet, brown	0.87
Bitumen-covered roofing sheet, green	0.86
Bituminous felt	0.88
Bituminous felt, with aluminized surface	0.40
Slate, silver grey	0.79
Slate, blue grey	0.87
Slate, greenish grey, rough	0.88
Slate, dark grey, smooth	0.89
Slate, dark grey, rough	0.90
Tile, clay, machine made, red	0.64
Tile, clay, machine made, dark purple	0.81
Tile, clay, hand made, red	0.60
Tile, clay, hand made, reddish brown	0.69
Tile, concrete, uncolored	0.65
Tile, concrete, brown	0.85
Tile, concrete, black	0.91

Where specific material is not mentioned above, an approximate value may be assigned by use of the following rough color guide.

1.—For white, smooth surfaces, use	0.25 to 0.40
2.—For grey to dark grey, use	0.40 to 0.50
3.—For green, red and brown, use	0.50 to 0.70
4.—For dark brown to blue, use	0.70 to 0.80
5.—For dark blue to black, use	0.80 to 0.90

(Figures are expressed as decimals of the intensity of solar radiation striking the surface)

they represent the best information obtainable for values of b in equation (4). Values for enough materials are included to permit selection of a suitable one to fit many cases in practice. The tabulated values show clearly the effect of color and of roughness of surface, with the factor increasing in value as the color approaches black.

Amount of Solar Energy Conducted

Turning attention to suitable values for g in equation (4) we find but little experimental work done and no comprehensive statement of the principles on which these values rest. Several facts about the values can be predicted, however.

In the first place it is evident that when solar radiation is transformed into sensible heat on the outside surface of a wall or roof, it must pass through the construction by conduction. We know that it will meet with increasing resistance to flow as the heat insulating properties of the wall or roof increase. Thus if a wall or roof were a perfect heat insulator none of the sun heat could pass through. There must therefore be some relation between the value of the coefficient U of a wall or roof[2] and the sun heat which can pass through.

[2]For values of this coefficient for most of the constructions now in use, see H. & V.'s Reference Data Sheets 71-78.

It is also known that with the heat lag of the materials commonly used in walls and roofs and with the intermittent character of the solar radiation usually received, the percentage of the solar heat which finally reaches the inside of the building to add to the cooling load is small. Therefore we would expect that values of g would be small numerically. What small amount of experimental evidence there is bears out this idea.

Table 3 has been constructed by making use of the assumption that there is a relation between the coefficient of heat transmission U and the factor g. The relationship is fixed by plotting points gleaned from the results of several reported tests and using equation (4) with g as the unknown to be solved for. The values from the curve are tabulated as Table 3. It is believed

TABLE 3—VALUES OF g FOR WALLS AND ROOFS

Coefficient U for Wall or Roof	Decimal Part of Absorbed Solar Radiation Transmitted to Inside
0.10	0.02
0.15	0.03
0.20	0.04
0.25	0.05
0.30	0.062
0.35	0.075
0.40	0.085
0.45	0.095
0.50	0.105
0.60	0.13
0.70	0.15

that they are the best values now available, although they are purely tentative.

Values for the area, the factor A in equation (4), are obtainable by measurement but it should be noted that while the area of a roof or wall is easily determined it does not follow that *all* of the areas will be affected by the sun. Selection of the areas to be used, especially the areas on which the sun shines at the time of the peak load, may require exercise of considerable judgment. Common sense methods are more helpful in making such a selection, though, than would any rules which might be stated.

What Should be Done

Enough data have now been presented to permit making an estimate of the sun heat through a wall or roof using equation (4). Details of how to apply this to a particular estimate are not yet agreed upon. Estimates of sun heat are still new enough so there is a divergence of opinion as to what should be done.

Examination of Table 1 shows clearly that the west, southwest, and northwest walls are the ones where important amounts of sun radiation are received in the afternoon. Moreover, the amount received at 4 p.m. is the maximum. In the case of flat roofs the maximum amount indicated is at noon. It is not at all clear that these separate amounts should be allowed for in the estimate, especially as they occur at different hours. Where a proposed installation requires close estimating it may be good practice to allow for the sun heat at all hours. The same is true where zoning is proposed and the changing loads at various hours are desired. A table like Table 1 permits the estimate being made at as many or as few hours as desired and permits making allowance for sun heat on all walls and roof or on as few as desired.

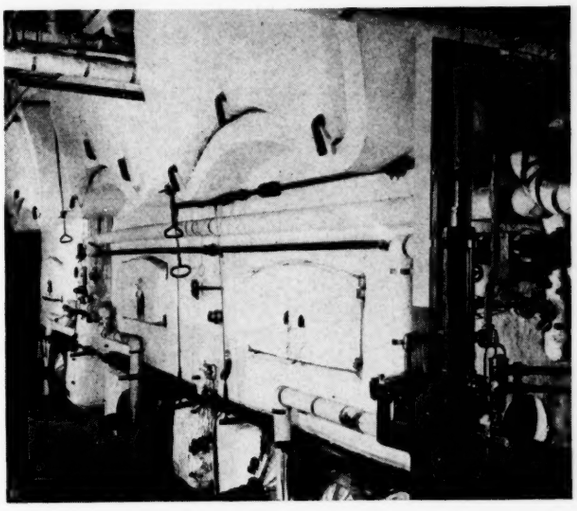
On the whole it cannot be said that the data or methods for estimating sun heat through walls and roofs are yet in good shape. Most of the weaknesses have been indicated. As in making estimates of the other heat quantities, however, existing data permit securing estimates of value when used with a proper element of judgment.

Hotel Piping Insulation Pays for Itself in 12 Months

EVIDENCE that boiler and piping insulation pays its way as a modernization project is found in the case of the Park Central Hotel, New York, which had some $2,000 worth of rockwool covering installed last year and saved more than $1,400 in fuel oil in the first 12 months of service.

One of the first things the insulation contractor did was to remove the old insulation. Then he covered the boiler doors with a 2-in. rockwool blanket, finished off with ½ in. of asbestos cement. The six boiler drums then were covered with a 1½-in. blanket of rockwool with the same finish. Fittings on a 90-lb. steam header over the boilers and on a 40-lb. steam line were covered. Altogether more than 4,000 sq. ft. in flat surface equivalent received new insulation.

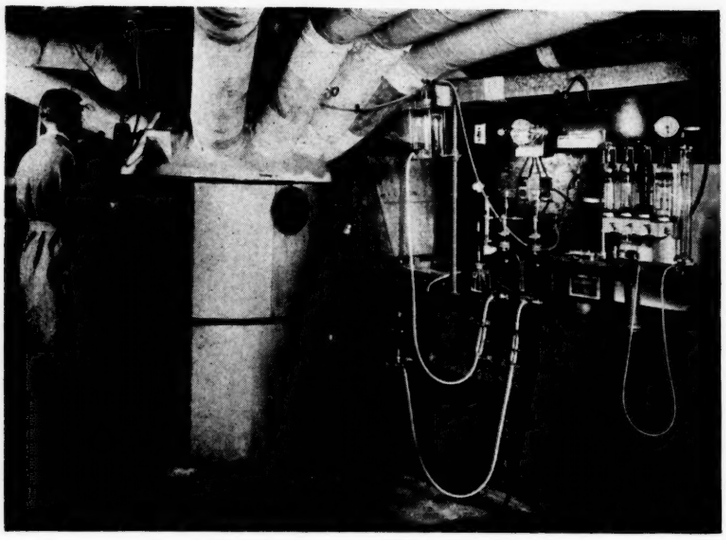
Insulating Products Corp., New York, which secured the contract, used the survey method to obtain the order. This is a regular feature of this company's selling methods. Salesmen solicit opportunities to survey, tabulate details of conditions found on prepared forms, make up their estimates, and tender them to building managers. Many times results must be guaranteed. It is essential in this sort of merchandising, states an executive of the contractor, to be conservative when making guarantees.



Partial view of the boiler and piping insulation job in the Park Central Hotel.

In the Park Central job, the estimated heat loss at the time of the survey was 978,240 B.t.u. per hr. The contractor estimated that the loss would be reduced to 97,924 B.t.u. per hr. following application of its material in the manner indicated. It was estimated that fuel saving in gallons of three cent oil would be $1,108 for 5,500 heating hours. In operation it was found that the saving exceeded this by $300.

Thus the customer, who intended to pay for the modernization from current savings over an 18-month period, discounted the bill at the end of 12 months when it was evident that the terms of the contract would be fulfilled. In summary, the improvement started to pay dividends to the hotel at or somewhat prior to the end of the 18 months and actually the modernization cost the customer nothing at all because the modernization paid for itself from savings.



Testing coal-fired furnaces at Battelle.

Stoker Applications in Warm Air Furnaces†

are studied by Battelle, where investigations are under way on effect of coal sizes and chemical treatment. The author suggests that furnace designers undertake research on the relation of their product to stokers

By RALPH A. SHERMAN‡

APPRECIATING the facts brought out as to the lack of development of satisfactory methods for burning coal in small installations the work of the first year at Battelle Memorial Institute is devoted largely to this phase of coal utilization.

Out of the west not many years ago came a development that is of great importance to the coal and heating industries—the small underfeed stoker. In 1933, 21,000 were sold; in 1934, 33,000; in 1935, about 45,000, and their expectations are for a sale of 70,000 to 75,000 in 1936. The figures are for the small stokers, a feeding capacity of 100 lb. per hr. or less, which account for about 80% of all stokers sold.

The annual tonnage of coal consumed by each of these stokers is not great but the aggregate tonnage is now appreciable and every producer and distributor of coal is confronted with the problem of the proper type and size of coal to market for their use. Because of their widespread use the satisfactory solution of this problem is of even greater importance to the industry than would be indicated by the relative tonnage; a high degree of satisfaction in automatic heating with low cost fuel will strongly influence public opinion in favor of coal.

One of the major problems, therefore, on which we are working at Battelle is that of the relation of the characteristics of coals to their performance on small underfeed stokers. We propose to take up each of the characteristics, as volatile matter, ash content, clinkering tendency, and size, by itself, as far as possible, and determine its relation to the stoker performance.

Because of its great importance the first characteristic of coal that we are studying is that of size. Burning tests are now under way on different sizes of representative coals. The results of these tests will not be complete for some time but some preliminary tests in connection with the size problem have been made the basis of Technical Report No. 1 of the Corporation.[1]

As it is obvious that some change of size occurs as the coal is fed through the worm from the hopper to the retort, we set out to determine the extent of this change in size that we might properly correlate the size of the coal fed to the hopper to its burning characteristics.

Among other things we found that the top sizes of coal, that is, from ¾ in. to 1½ in., were so largely crushed that very little reached the retort to be burned. From this standpoint, therefore, there is no purpose in using these large sizes of coal. It was also found that there is considerable segregation of different sizes of coal in the retort of the stokers. The fines go to the side of the retort opposite the hopper and the coarse sizes concentrate on the near side; this tends to result in uneven burning. Of all the sizes tested, the — ¾ in. + 0 coal showed the least segregation. Oil treatment of coal such as is used to prevent dust was found not to affect the crushing of the coal but it did decrease the segregation. It is expected that this work will lead to a more complete knowledge of the effect of size so that the producer and dealer will not have to prepare and carry so many sizes of coal.

In connection with this study of small stokers we are conducting, as the contribution of Battelle Memorial Institute, a fundamental study of the process of combustion in underfeed fuel beds. This should throw light on a process yet poorly understood and may lead to advances in stoker designs.

During the present heating season we are making a survey of heating results in a number of homes in Columbus using various kinds of coal, both hand and stoker-fired, coke, oil, and gas in steam and hot water boilers and in warm air furnaces. The heating plant is fired as usual by the occupants of the house and we have men on duty 16 hr. a day to keep records of the attention to the fire, the amount of fuel fired, the drafts, the temperatures, pressures, smoke, and flue gas composition; a recorder gives us the temperature and hu-

[1]This refers to Bituminous Coal Research, Inc., a non-profit making subsidiary of the National Coal Association, organized to study coal utilization and other problems relating to the soft coal industry, and sponsor of the research at Battelle.

†Abstracted from a paper presented at the 23rd annual convention of the National Warm Air Heating and Air Conditioning Association, January 30, 1936.

‡Battelle Memorial Institute, Columbus, Ohio.

midity in the living quarters. Data are taken over a period of five days in each house. From the data obtained we can calculate the thermal efficiency of the unit and the distribution of the heat lost. The accuracy obtained can not equal that of a laboratory set-up but it should give us a picture of what happens in the actual home. In one house where coke was hand-fired, the efficiency of heat utilization was 78% and the unaccounted for losses were only 8%.

As most small underfeed stokers are designed for the removal of the ash as clinker, we have already made an investigation and submitted a report on the possibility of adding chemicals to induce clinkering with those coals which do not readily form a clinker, particularly in mild weather when burned at low rates.

To realize the full possibilities of the combination of the automatic stoker and the warm air furnace, however, the furnace industry, individually or as a group, should undertake research on the design of furnaces for stokers. Stokers can work well in furnaces designed for hand firing but improper installation can greatly decrease the level of performance that should be attained. The function of the stoker is to feed and burn the coal to liberate its energy as heat. The function of the furnace is to transfer that energy to the air to be circulated through the space to be heated. The method of heat transfer from a stoker fuel bed is somewhat different to that in a hand-fired fuel bed. The furnace will perform better if it is designed to fit the stoker.

Other mechanical problems in the application of stokers to warm air furnaces need to be solved. With the hopper-feed stoker it is desirable to place the stoker at one side of the door to allow complete freedom in removal of clinker. This is ordinarily entirely feasible with boilers but it is not with warm air furnaces because of the difficulty in preventing the transmission of vibrations of the stoker to the metal casing.

Ice for Air Conditioning Motor Buses

THERE is an intense interest in the subject of ice for motor bus cooling. Ice can do the job, I believe, better than anything else, but there will be plenty of competition in the field.

Bus air conditioning with ice was tried by several bus companies last summer, with various degrees of success. Among the companies trying it were the Eastern Greyhound Lines and the Edward Lake to Sea Lines. Much good experience was obtained by these companies and I believe they will continue with ice in an experimental way until they have the job satisfactorily solved to their own satisfaction.

There may be some misconception about the use of ice in air conditioning. So many think that apparatus using ice for air conditioning is somewhat of a makeshift—that it cannot do the job as well as something else of a mechanical nature. This is not the case. Ice, in a well designed system, can be complete in every sense. The system using ice as a refrigerating element is usually referred to as "the ice activated system," and the ice is used to create a supply of cold water; the cold water, in turn, being used to cool and dehumidify the air, either by means of a water spray or coil (radiator), usually of the finned tubular type.

During the past summer there was some interesting experimental work done with ice activated bus cooling and some trouble was encountered with the ice bunker itself because it was improperly insulated and the construction too flimsy. Of course, it is very important that this bunker should be well insulated against heat leakage through the walls and rigid and sturdy in construction, with special attention given to door and hardware design, for these parts are exposed to rather rough usage when the bunker is loaded.

The weight added to the bus with this system may at first seem excessive, and if the bunker were carried full of ice at all times, this would be a serious consideration. But the bunker is not filled with ice at all times and the average weight is surprisingly low.

Some bus operators are thinking along the line of providing cooling only when the bus is moving at a slow speed through large cities, or standing idle in terminals. If this is practical, and also if the actual number of days when cooling is required are few, then the average added weight of an ice activated air conditioning system on a bus is surprisingly low.

It is generally admitted among engineers that an ice activated air conditioning system shows up most favorably from a "cost of operation" standpoint on low load factor cooling jobs, meaning by "load factor" the proportion of cooling time to the total elapsed time; for example:

If the cooling season is considered to be May 15 to September 15, a total of 123 days, or 2,952 hr., and if, by the careful operation of a cooling system, it is found that actual cooling is required for only 300 hr., during this period, the seasonal load factor is said to be 9.8%, or on a yearly basis, the annual load factor 2.9%, meaning that during the cooling season actual cooling was needed 9.8% of the season's time, or 2.9% of the annual time.

The most important advantage is the minimum of additional power required to operate the system. With modern bus engines there is probably sufficient excess mechanical power available to create the necessary cooling by mechanical means, but the conversion of the mechanical energy into cooling energy is a difficult matter to accomplish. The ice activated system, requiring only power to operate a small water circulating pump assuming that power is now already provided for ventilation, establishes a minimum drain on the present power plant.—*Abstracted from a speech by Wm. A. Schmid, Jr., at annual meeting of Motor Bus Operators.*

Summer Air Conditioning in 1935 Up 28.6% Over 1934

FINAL figures on summer air conditioning in 1935 as prepared by important utility companies are now available and shown here in Tables 1 and 1A. As these utility companies serve customers in metropolitan areas having a large part of the total urban pupulation, these figures come close to being totals for the whole country.

The tables are so arranged that they include identical cities on which HEATING & VENTILATING has published similar tables for previous years. This permits making direct comparisons from year to year. From such comparisons of yearly figures an interesting record of trends of the volume of business can be prepared.

To keep clear what is included in each of the columns of Tables 1 and 1A, the following classifications of building types are given:

Commercial: All department stores; retail commodity stores, such as shoes, clothing, groceries, novelties, food, and other similar retail stores; retail personal service stores, such as beauty shops, barber shops, funeral homes, and similar places; eating and drinking places, such as restaurants (including those in hotels), taprooms and similar places but excluding dance halls, night clubs, and similar recreational establishments;

TABLE 1.—NUMBER AND SIZE OF SUMMER AIR CONDITIONING INSTALLATIONS IN TYPICAL CITIES DURING 1935
(As reported by 27 utilities to whose lines the installations are connected)

City	Commercial		Industrial		Theater and Recreational		Public and Institutional		Residential		Total	
	No.	Hp.	No.	Hp.	No.	Hp.	No.	Hp.	No.	Hp.	No.	Hp.
Boston	38	1,401½	6	401	0	0	0	0	10	49	54	1,851½
Metropolitan New York:												
Manhattan and Bronx	181	6,031½	6	53¾	19	2,160	0	0	4	10¼	210	8,255½
Brooklyn	41	2,054¾	5	325	10	225½	0	0	8	9	64	2,614¼
Queens	3	43¼	1	36¼	4	80	0	0	1	1¾	9	161¼
Newark	14	325	1	40	3	53	3	90	0	0	21	508
Jersey City	4	4	3	30	0	0	0	0	0	0	7	34
Philadelphia	61	3,544½	17	685¾	11	431½	3	47	24	61¾	116	4,770½
Baltimore	63	1,615	2	110	11	633	3	25	9	17	88	2,400
Washington	94	3,034	0	0	6	531	23	5,003¼	36	82¾	159	8,634
Rochester	18	524	2	126	0	0	0	0	8	18¼	28	668¼
Buffalo	No Detailed Report										46	656
Pittsburgh	65	1,218¼	0	0	2	170	0	0	28	34½	95	1,422¾
Toledo	20	338½	3	87½	1	60	0	0	2	4	26	490
Detroit	43	2,450¼	28	1,862½	1	5	4	50¾	14	27	90	4,395½
Cincinnati	60	2,154	0	0	4	303	0	0	32	48	96	2,505
Indianapolis	25	1,318	2	12	0	0	0	0	0	0	27	1,330
Metropolitan Chicago:												
Chicago	234	3,213¾	20	499¼	31	1,665	8	56¾	24	26¼	317	5,461
Environs	No Report											
Milwaukee	12	288¼	4	51½	6	93½	0	0	3	7¾	25	441
St. Louis	101	2,577¾	13	488½	5	2,158¾	0	0	68	145¾	187	5,370¾
Minneapolis	23	266	0	0	4	70	1	100	7	23	35	459
St. Paul	15	117	0	0	0	0	1	20	3	12	19	149
Denver	13	84½	1	1	1	9½	0	0	34	34	49	129
Atlanta	No Report											
Birmingham	18	934	0	0	1	13	0	0	3	4½	22	951½
Louisville	29	667	0	0	2	45	0	0	4	1	35	713
New Orleans	33	1,901½	1	8½	1	24	1	55	23	78	59	2,067
Dallas	43	1,029¾	0	0	2	107½	1	11¾	18	57¼	64	1,206¼
Houston	31	2,133½	0	0	2	283½	1	18	12	31¼	46	2,466¼
Total	1,282	39,269½	115	4,818½	127	9,121¾	49	5,477½	375	784	1,994	60,110¼

TABLE 1A.—NUMBER AND SIZE OF SUMMER AIR CONDITIONING INSTALLATIONS IN SEVERAL ADDITIONAL CITIES DURING 1935

City	Commercial		Industrial		Theater and Recreational		Public and Institutional		Residential		Total	
	No.	Hp.	No.	Hp.	No.	Hp.	No.	Hp.	No.	Hp.	No.	Hp.
Miami	32	596½	0	0	3	532¾	2	43¾	8	20	45	930¼
Akron	7	45½	0	0	0	0	0	0	3	3½	10	49
Kansas City	70	1,406	0	0	1	45	7	2,147	38	123½	116	3,721½
Syracuse (11 Mo.)	No Detailed Report										6	211
Hartford (11 Mo.)	No Detailed Report										14	172
Scranton (10 Mo.)	No Detailed Report										10	138
Richmond	No Detailed Report										74	1,867
Omaha (11 Mo.)	No Detailed Report										45	1,133
San Francisco and No. California (11 Mo.)	No Detailed Report										164	809
Total	No Detailed Report										484	9,030¾

TABLE 2.—SUMMER AIR CONDITIONING, 1933-35, AS REPORTED BY 27 LARGE UTILITIES.

	Commercial		Industrial		Theater and Recreational		Public and Institutional		Residential		Total	
	No.	Hp.	No.	Hp.	No.	Hp.	No.	Hp.	No.	Hp.	No.	Hp.
Prior to 1933	599	54,022	164	37,400¾	304	69,296¼	20	1,117	94	289	1,161	162,125
1933	516	13,101	45	1,641¾	48	3,393	23	491½	167	312	799	18,939¼
1934	1,000	25,091¼	63	1,675¼	96	4,295	68	15,145	341	667¾	1,568	46,874¼
1935	1,282	39,269½	115	4,818½	127	9,121¾	49	5,477½	375	784	1,994	60,110¼
Total, Jan. 1, 1936...	3,377	131,483¾	387	45,536¼	575	86,106	160	22,631	977	2,052¾	5,522	288,048¾

office buildings; private and general offices and board rooms in commercial and private office buildings, including banks and similar financial buildings; hotels, clubs, and apartments, including guest rooms therein.

Industrial: Installations made to further some process or to insure employe comfort in industrial buildings. Where private offices, general offices or board rooms are located in buildings close to or part of a factory, they are included as industrial.

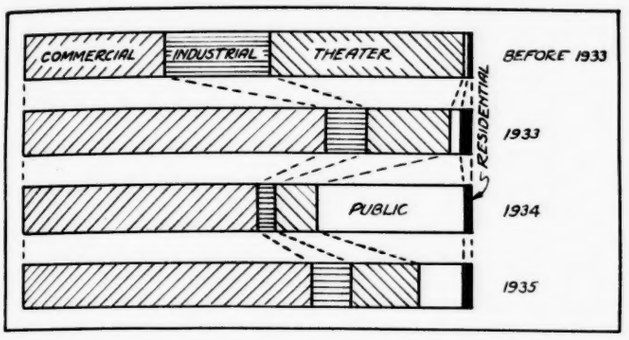


Fig. 1. Apportionment of summer air conditioning among different classes of installations, by years.

Theater and Recreational: Theaters, including movie and legitimate show houses, auditoriums for recreational use, night clubs, dance halls, and installations where amusement or recreation is the prime function.

Public and Institutional: Public buildings for whatever purpose used, including office space, court rooms, legislative halls, record storage, libraries, publicly-owned auditoriums, public and private hospitals, and institutions of all kinds.

Residential: This really implies residences, for hotels, clubs, and apartments are included elsewhere. In the main the residences reported are of the detached one-family type.

Table 2 shows the total number of installations and total horsepower installed in identical cities in each of the building classifications during recent years.

There is thus a definite record of nearly 300,000 hp. and upward of 6,000 separate installations of summer air conditioning in buildings up to January 1, 1936. The actual total is still not known but it is believed that the total number is around 8,000 and that the total horsepower exceeds 350,000. Dollar value at retail is estimated at from $100,000,000 to $120,000,000.

1935 a Good Year

That 1935 was a good year comparatively can be seen from Table 2 by noting that the horsepower figures reported for 1935 are nearly equal to the sum of those for 1933 and 1934. In other words, 1935 brought nearly as much volume as did 1933 and 1934 combined. Another way of expressing the same thought is to compare 1935 with 1934 alone. When this is done it will be seen that there was 28% more volume as shown by Table 2.

Installations in commercial buildings dominated the picture during 1935 as during other recent years. Fig. 1 shows how the volume has varied recently in these identical cities. In 1935, 65% of all the dollars spent by owners for summer air conditioning went to pay for installations put into buildings used for commercial purposes. In 1934, the volume in public building work reached its height due to the immense cooling plants put into the government buildings at Washington. In 1935 the total from this source fell sharply. Installations in industrial plants picked up after decreasing steadily during preceding years. Theaters and recreational buildings showed the same tendency. Installations in residences are shown to represent but trifling percentages of the total either in 1935 or preceding years.

In the January, 1936, issue of Heating & Ventilating a chart was published showing a 10-year record of summer air conditioning. This was based on reports for 11 months of 1935 from several cities. With complete results now available this chart has been revised to take the changes into account and is reproduced here as Fig. 2.

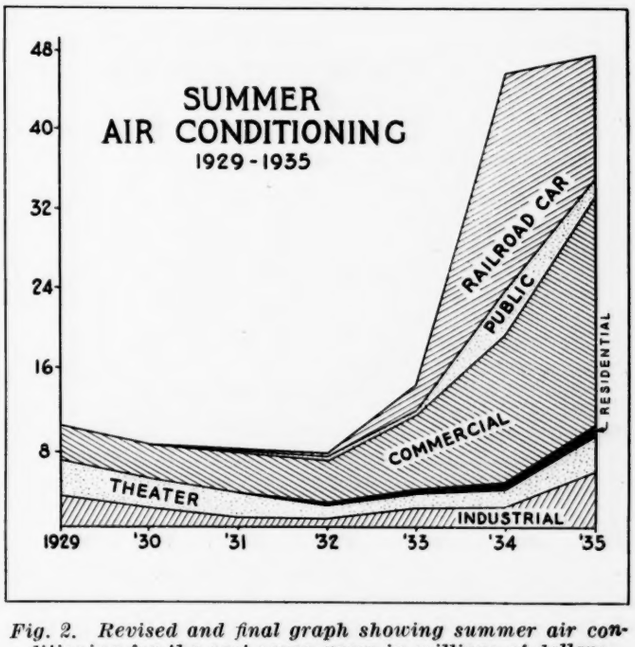


Fig. 2. Revised and final graph showing summer air conditioning for the past seven years in millions of dollars.

The Influence of Counterflow on Convector Capacity

By M. WILLIAM EHRLICH†

USUAL steam flow formulas for determining the carrying capacities of pipes, or for capacity ratings of radiators and convectors, do not take a counterflow into consideration. Such computations are based entirely on a one-way flow, in which the steam and condensate travel together in a uniform direction. This form of one way circulation within the pipe or heater is here designated by the term *uniflow*.

The word *counterflow* is used to describe the other form of circulation, in which steam and condensation travel against each other, in opposite directions but within the same pipe or heater. There are few published data on counterflow, although the generally accepted tabulations which give pipe sizes for steam heating do serve as a guide, and it is this information which has been used in a study to find the relationship existing between uniflow and counterflow circulation. The data on pipe sizing are based on research and the results appear to be definite for given conditions. Applying these data to the type of convector which consists of horizontal pipes, it can be shown that performance with counterflow operation must be less than with uniflow circulation.

In the logarithmic chart, Fig. 1, is shown the relationship between capacity, pipe size, and velocity, for both uniflow and for counterflow in pipes of various sizes. These are maximum values and should not be exceeded for safe practice in single pipe work. Fig. 1 is based on research of pipe capacities and is made from Table 7 of the 1934 ASHVE Guide. The figures apply to a 1-oz. pressure drop and are for horizontal

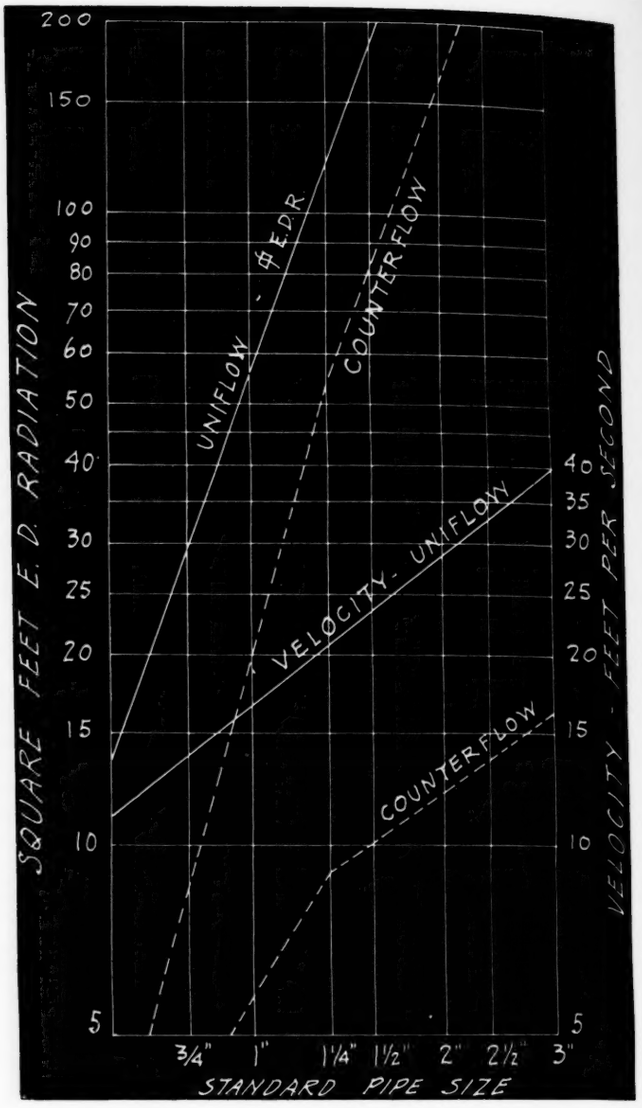


Fig. 1. Relationship of size, steam velocity, and capacity of pipes with one-way flow (uniflow) and counterflow.

mains for two-pipe systems and for radiator and riser run-outs for one-pipe systems.

For example, consider a ¾-in. pipe. At uniflow, this will carry 30 sq. ft. of radiator at a velocity of 14 ft. per sec. At counterflow, the same size would carry so little and the velocity would have to be so low, that the ¾-in. size is not recommended in one-pipe steam heating service.

Then consider a 1¼-in pipe. The capacity of this pipe at uniflow is 122 sq. ft. of radiator, and its velocity is 20½ ft. per sec. At counterflow, the same size pipe would carry only 55 sq. ft. of radiator since its velocity is only 9 ft. per sec. These figures are definite for the conditions of pressure drop under which the tests were run.

A similar relationship is shown for any other pipe size as given in Fig. 1. A higher velocity is possible with uniflow conditions than is possible with counterflow, for a higher velocity is possible at uniflow without causing noise for prevention of circulation. Thus, other things being equal, the velocity rate may be used as an index of performance.

This idea is extended in the logarithmic chart, Fig. 2,

†Chief Engineer, Commodore Heaters Corp., New York.

where *relative* values are plotted against pipe areas. Taking uniflow performance as the base, then relative results at counterflow are established for the same conditions. The solid line curve indicates the influence on velocity as taken from Fig. 1. (When the peak has been reached, there is a change in rate, and so the relative capacity is lowered. But if this maximum point were maintained, then the curve would have the form shown by the dotted line, and here this is taken as the preferred position.)

To use, follow up from the size scale to the curve, and from this intersection, read across to the scale at the right. For 1-in. pipe, it is shown that the counterflow capacity is about 0.37 of the uniflow quantity. Again, a reading for pipes from 1¼-in. to 3-in. pipe shows that the greatest value at counterflow is 0.46 of the uniflow capacity.

Another picture of this relationship is afforded by the left-hand scale of Fig. 2. This shows the relative velocity that would be needed at counterflow to produce the same capacity obtainable at uniflow with the same size of pipe. Using the same examples as before, the 1-in. pipe shows a relative value of 2.7 and the larger pipes to 3-in. size require 2.2 times the velocity. Since the relative value of 1 is unity at counterflow, any number above 1 shows that the safe practical velocity will be exceeded that number of times, and if used would therefore result in faulty and troublesome circulation. Therefore the safe carrying load at counterflow is limited to the values indicated by the relative capacity scale.

Effect on Heater Capacity

If the same reasoning is applicable to the performance of radiators and convectors, then the conclusion must be that the heating output is less at counterflow than at uniflow.

Some heating units have vertical columns or tubes and may therefore operate at higher velocities than heaters with horizontal sections or tubes, but the relative values for counterflow and uniflow would probably remain about the same. In respect to pipe lines, undoubtedly all heaters may have a higher velocity through them, but the maximum is reached at rating.

Capacities of the heating output of radiators and convectors are established by tests under uniflow circulation. When any such heater is given a rating of, say, 10 sq. ft. it means that the capacity at uniflow is 10 sq. ft. E.D.R. From the foregoing reasoning, however, it follows that its capacity at counterflow would be less.

On this premise, the only real question then would be—"How much less?" To determine the magnitude of the relative value Fig. 2 may be applied to arrive at some indication of the relative results.

Assume a convector in which the capacity determination showed a developed uniflow velocity of 30 ft. per sec. through an area of 0.7 sq. in. By reading the chart it is found that at counterflow the relative capacity would be 0.33 of the rating, or that the relative velocity to give full rating is three times the safe velocity at counterflow.

As another example, take a convector with larger steam ways. The rating determination showed 20 ft. per sec. as the uniflow velocity through an area of 2 sq. in. Now, reading the chart for this condition, it is found that the relative capacity is 0.46 at counterflow; or that the relative velocity is 2.2 times the permissible velocity, to give full rating capacity at counterflow.

As the velocity is the practical limiting factor, and as the counterflow rate is consistently at a lesser speed than at uniflow, it should follow that any radiator or convector must have two distinct heat outputs, i.e., the one for rating at uniflow, and the other, a lower capacity, at counterflow.

The material of which the heater is made, its form of heat emission, or its internal volume, all may have some bearing on its functioning, but for any given radiator or convector it is the circulation—uniflow or counterflow—that determines the final performance and operating result. Therefore, it cannot be right to designate the same heater with a single rating for such two differently distinct service conditions.

In current practices, arguments have been advanced that greater volume within a heater would overcome

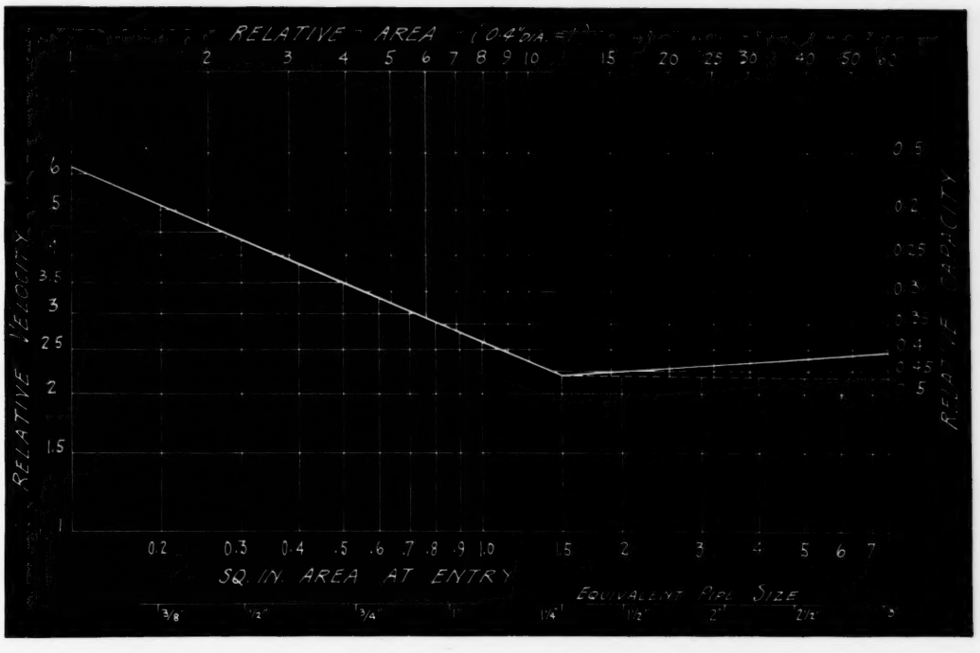


Fig. 2. Relative velocities and capacities of pipes with uniflow compared with counterflow.

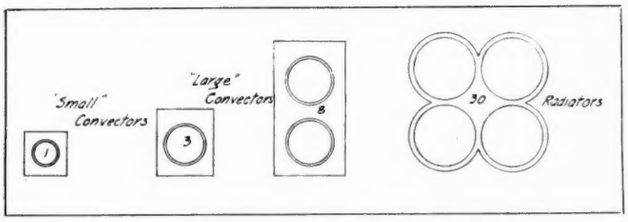


Fig. 3. Relative cross-sectional areas of various convectors.

the differences in the characteristics or the velocity requirements between counterflow and uniflow. However, volume is only a measure of internal cubic contents. A heater of large volume may even have smaller steam passages or waterways than one of smaller volume, depending upon individual features of design. Thus, the area through which steam has to enter and the water come out is the determining value in this respect. To see how this lines up, relative areas are shown in Fig. 3. The base used as unity or 1 is an inside diameter of 0.4 in., so that the horizontal small-tube convector may be included in these considerations. The larger heaters also shown are drawn to the same scale.

For purposes of example, an average size convector has a relative area of 3. Refer to this on the top scale of Fig. 2, read down to the curve, and from the intersection, read at right or left for the answer. Thus, the counterflow capacity is shown to be 0.25 of the uniflow rating, while the relative velocity needed for full capacity is 3.9 times greater than available at counterflow. Or consider a radiator with a relative area of 40. A similar relationship is shown to exist, although the values are different. For this condition the counterflow capacity is shown to be 0.46 of the uniflow rating, while the velocity needed to give full capacity is 2.2 times that of the safe velocity at counterflow.

Some Velocity Figures

The information on piping has been given in definite numerical values. The considerations on radiators and convectors being based on these data have been treated so far in terms of relative values. But these may be converted into actual figures by applying the relative values to definite numbers for known conditions.

In the vertical column radiator the velocity rate is said to be 18 ft. per sec. An average case of the horizontal tube convector may have a velocity rate of 30 ft. per sec. Each individual type of heater will show its specific velocity resulting from tests at uniflow, when the capacity rating is being determined, thus making definite values available for individual heaters. The range, however, will probably be higher than for the pipes shown in Fig. 1. In this diagram the highest points show safe limits in velocities as 17 ft. for counterflow, and as 40 ft. per sec. for uniflow. On the average, therefore, it would not be unfair to assume, for purposes of illustration, that a rate of travel at 15 ft per sec. is practicable in counterflow for both radiators and for convectors.

Applying this to the heaters previously cited, we have, for the convector, 15 x 3.9 or 58.5 ft., and for the radiator, 15 x 2.2, or 33 ft. per sec. These then are the hypothetical velocities needed at counterflow (in these two cases) in order that the heaters may deliver the full capacities equal to their ratings at uniflow.

It is unreasonable to expect water of condensation to flow counter to the onrush of steam that may have a force equal to the intensity of a gale. The velocity washes the water against the air valve, tends to hold it there, and so prevents complete circulation, so that the uniflow capacity rating assigned to the heater cannot be reliably delivered, and there is consequent trouble in operation and failure to meet the specified heating results.

Southern Pacific to Spend $1 Million on Air Conditioning

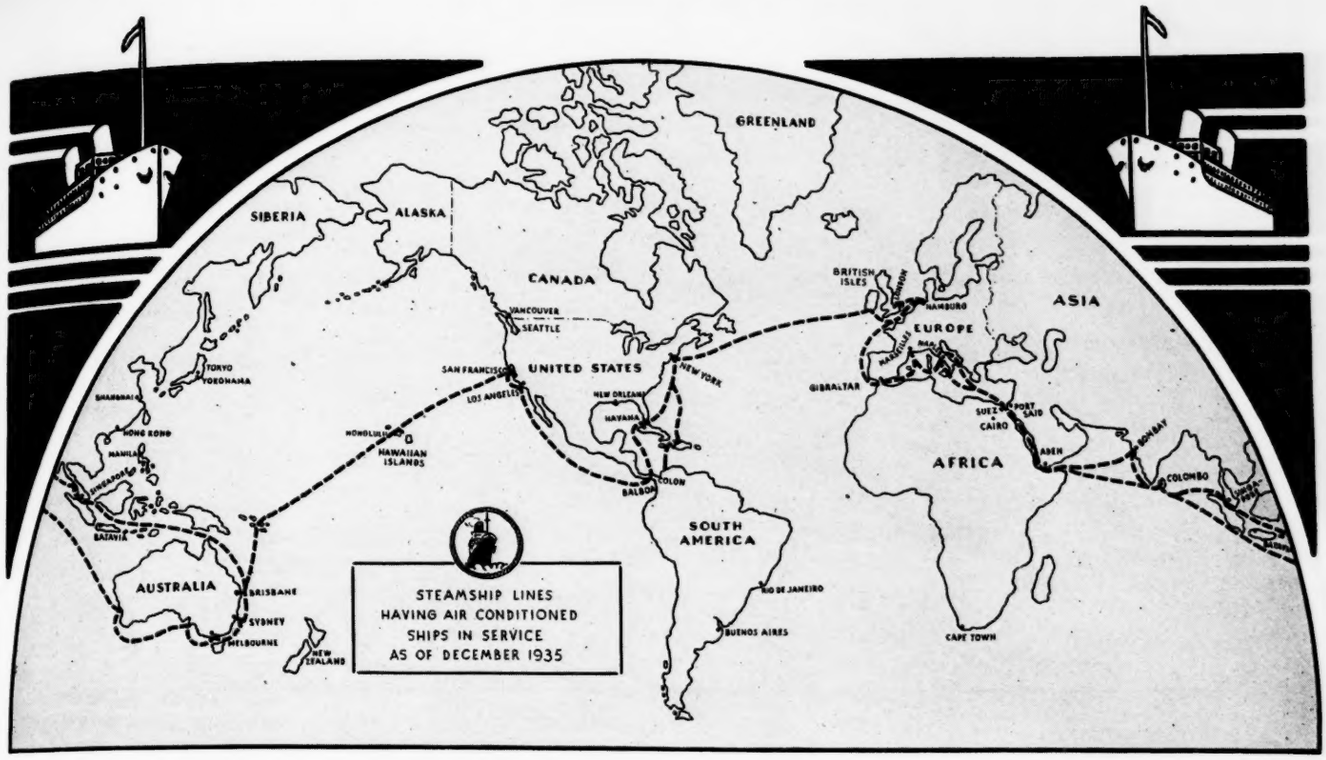
IN its air conditioning program for 1936, the Southern Pacific Company will spend $1,026,719 on its Pacific Lines and the lines in Texas and Louisiana. Of this amount, $620,081 is for air conditioning of the company's own cars; $380,698 to modernize and improve passenger cars, and $25,940 for electrical facilities at terminals for air conditioning operations.

In additional to the company's cars being air conditioned in its own shops, the Pullman Company will also air condition a total of 105 standard and 26 tourist cars for use on Southern Pacific Lines.

The completion of this program by early summer will give Southern Pacific a grand total of 588 air conditioned cars in operation on its lines between the Pacific Coast cities, Ogden, New Orleans, and Guadalajara, Mexico. Of this number, 256 will be the company's own cars, and 332 belong to the Pullman Company. The types of cars conditioned include chair cars, coaches, diners, and lounge cars.

It is interesting to note that three methods of air conditioning are used in the Southern Pacific company's cars—ice, steam ejector, and mechanical systems.

The ice activated system used in a majority of the company's cars is a development of the Southern Pacific's own mechanical department. The cooling medium is ice water secured by the melting of ice in a large tank under the car body. This ice water is pumped through pipes to air cooling coils in the roof of the car. Filtered fresh air to be circulated through the car is cooled to a comfortable degree by being forced over the outer surface of the air cooling coils. The process is continuous, water passing from the air cooling coils back to the ice tanks where its temperature is again reduced for recirculation.



Ocean Liners Turn to Comfort Cooling

ALTHOUGH comfort cooling was in use on shipboard before it proved successful in railway work, it has not had the spectacular advance which has so notably capped its introduction into the latter field. Space aboard a ship is always at a premium, and once in service the chances of adding additional equipment are small.

It is rather surprising, then, to learn that there are now in operation 14 passenger ships with comfort cooling. This, in spite of the fact that only a small number of ships have been built since 1929. Practically every one of the better passenger boats constructed during the past few years has had some cooling installed.

While it is true that in practically all the ships now in service the only passenger sections cooled are the dining rooms, it is at least a start and gives passengers an opportunity to become familiar with the benefits of cooling on shipboard. Some of the ships under construction at the present time planned to use cooling more extensively than present ships do. For instance, it is expected that the *Queen Mary*, Cunard White Star Limited's new superliner, will have its dining rooms, lounges, and some staterooms cooled.

The operations of these ships are not limited to one section of the globe but are fairly equally divided over all of the main ship lanes. It is interesting to note that the recent addition of two new liners on the London-Australia run now makes it possible to travel round the world on boats equipped with comfort cooling.

The map shows that there are three liners with comfort cooling equipment in service between San Francisco, Hawaii, and Australia; three between New York and San Francisco; two between New York, the West Indies, and South America; three on the North Atlantic lanes, and one in the Mediterranean between Italy and Egypt.

The Pennsylvania

The requirements for air conditioning on shipboard are in many instances much the same as in high class shore installations. The systems must be reliable, quiet, and safe. Unlike most shore installations, such systems must be designed to cope with an extremely wide variety of weather conditions.[1] For instance, the apparatus placed on the new Orient Liner *Orion* is designed to maintain comfortable conditions with outside temperatures ranging from 100° to 15° with relative humidities of 100% to 40%. In the tropics the system can reduce the temperature 10° to 15° and the relative hu-

[1]For a discussion of marine heating and ventilation practice see "Modern Practice in Marine Heating and Ventilating" by William J. Deed, HEATING & VENTILATING, October, 1931, page 50.

AIR CONDITIONED LINERS

(April, 1936)

PENNSYLVANIA	*Panama-Pacific Line*	*New York to California via Panama*
VIRGINIA	*Panama-Pacific Line*	*New York to California via Panama*
CALIFORNIA	*Panama-Pacific Line*	*New York to California via Panama*
HAITI	*Colombian Line*	*New York to West Indies and S. America*
COLOMBIA	*Colombian Line*	*New York to West Indies and S. America*
NORMANDIE	*French Line*	*Havre, France to England and New York*
MARIPOSA	*Matson Lines*	*San Francisco to Hawaii and Australia*
MONTEREY	*Matson Lines*	*San Francisco to Hawaii and Australia*
LURLINE	*Matson Lines*	*San Francisco to Hawaii*
MANHATTAN	*United States Lines*	*New York to England, France and Germany*
WASHINGTON	*United States Lines*	*New York to England, France and Germany*
VICTORIA	*Lloyd Triestino*	*Triest, Italy to Alexandria, Egypt*
ORION	*Orient Line*	*London to Australia via Italy and Ceylon*
STRATHMORE	*Peninsular and Oriental*	*London and Australia via Bombay*

midity from 100% to 55%. When the outside temperature is 15° it is possible to maintain a temperature of 67° with a relative humidity of 50%.

Undoubtedly, this represents the severest conditions which are likely to be encountered by any one ship, since this ship will be on the run from England to Australia. On this 13,000-mile trip all types of weather are encountered—from the cold of the wintery North Atlantic to the intense heat and stifling humidities of the Red Sea.

On ships designed for use on one particular run, it may be possible to plan for less severe conditions. For instance, a liner in the North Atlantic service will probably never encounter higher temperatures and humidities than exist in the New York harbor. The temperatures used in New York are 95° dry bulb and 75° wet bulb. For winter service the design temperatures will depend on what temperatures are encountered in the North Atlantic during the winter.

In designing the apparatus for marine use, allowances must be made for the rolling of the ship. This means that special baffles must be placed in the receiver of the spray chamber to prevent the water from sloshing around.

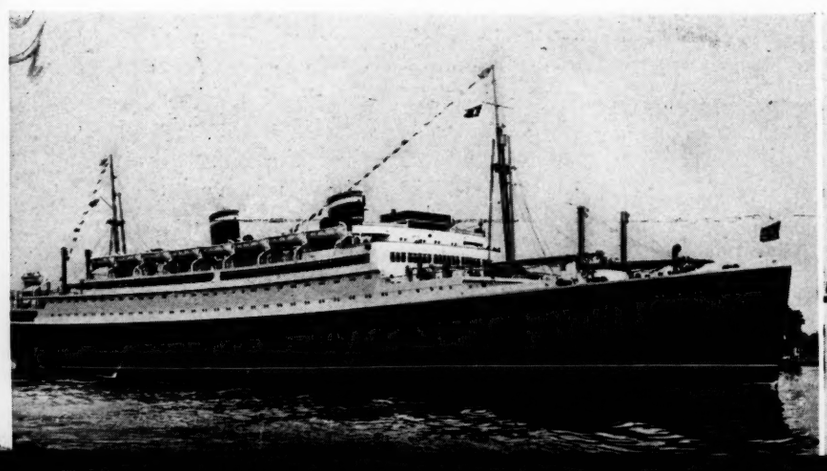
Cooling in the dining rooms, lounges, and other public sections of the ship makes it possible to increase considerably the revenue derived, for it allows these public rooms to be placed in the interior of the ship away from the outer and more desirable locations. Staterooms overlooking the sea are much in demand among travelers and consequently produce higher revenues.

When the public rooms are not conditioned they practically have to be placed in these more desirable locations. The French Line took this fact into consideration when its superliner *Normandie* was designed and placed the air cooled dining room in the center of the ship with staterooms on either side. This, in spite of the great size of the dining room which is 300 ft. long, 50 ft. wide, and 30 ft. high.

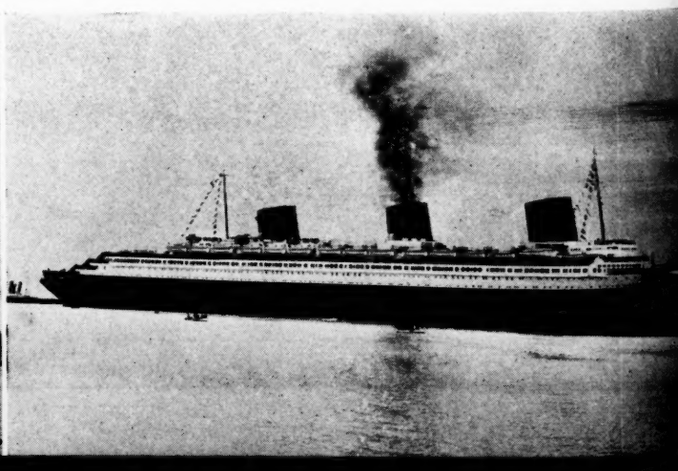
One of the newest passenger liners to be put into operation is the P. and O. *Strathmore,* which is at the present time on the London-Bombay, India, service. After May, however, the ship will operate between London and Brisbane, Australia, via Gibraltar, the Red Sea, India, Ceylon and Freemantle.

On this boat only the first class dining room is cooled. The ship also has a large amount of refrigerated space for perishable cargo. The refrigerating machinery is just forward of the boiler room and consists of four electrically driven horizontal twin compressors. Usually two of the machines will cool the provision room and insulated cargo space and one, the dining room. The fourth machine is for emergency use on either service. The machines are of the CO_2 type and are each driven by a 125-hp. variable speed motor. The

The Manhattan



The Normandie



working parts are all enclosed and forced lubrication is fitted to all the principal bearing surfaces.

The condenser coils are built up of copper tube and are enclosed in a circular cast iron housing. Two centrifugal pumps circulate sea water through the condensers and five centrifugal pumps circulate the brine.

The Orient liner *Orion,* also in the London-Australia service, has its first class dining room cooled. This liner, which is 665 ft. long, has accommodations for 1,139 passengers, and was launched last year.

A corner of the "Pennsylvania" dining salon

The equipment is placed amidships just above the dining salon dome. It consists of multivane centrifugal supply fan, marine type dehumidifier and humidifier, centrifugal spray pump, preheaters and afterheaters, and two mixing chambers with automatic mixing dampers. The control system is actuated by compressed air. Refrigeration for cooling and dehumidifying is obtained from a central CO_2 plant which also serves the refrigerated cargo spaces.

Perhaps the largest installation of air conditioning that has yet been made is the one aboard the *Normandie.* This system serves the first class dining room and eight small private dining rooms. Two central conditioners are employed for cooling and heating these rooms. One conditioner, located on the upper decks near the second stack, takes care of the front of the dining room and the other, located in the third stack, the rear of the dining room. One centrifugal compressor of 185 tons of ice melting capacity supplies both conditioners. This is located in the engine room.

The centrifugal compressor is driven by a 225-hp., d.c. variable speed motor through a step-up gear for increasing the speed up to 3600 r.p.m. Heat is removed from the condenser by circulating sea water through it. The maximum consumption of water is 4,240 cu. ft. per hr. at 81.5°. The refrigerant used is methylene chloride.

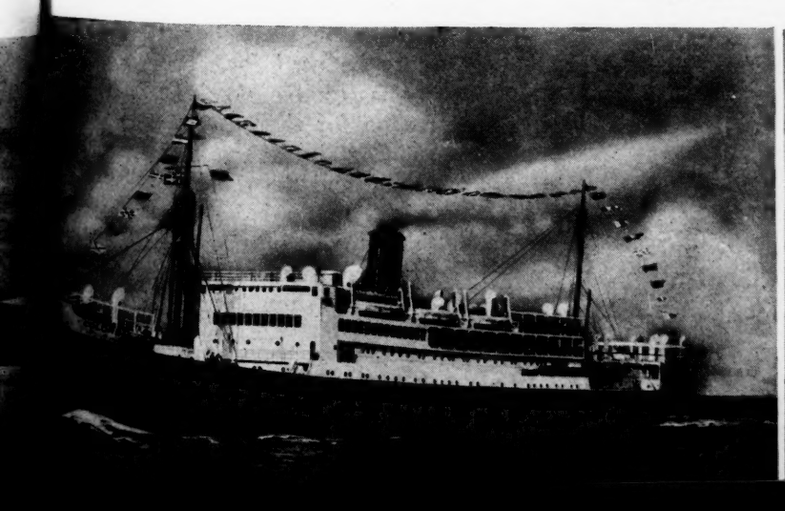
Approximately 16,000 cu. ft. of outside air are taken in every minute and mixed with 42,000 cu. ft. of air withdrawn from the conditioned rooms through openings in the panelling near the lower portion of the wall. The air passes through a mixing chamber which proportions the outside and recirculated air; then through a battery of preheaters which are used only in the winter to bring the temperature of the air up to a point where it can absorb sufficient water to obtain the desired humidity. In the spray chamber the air is cooled and dehumidified during the summer and humidified during the winter. Reheaters are used only in the winter. A large slow speed centrifugal fan draws the air through the reheaters and forces it through the distribution ductwork to the outlets in the upper part of the walls of the rooms. Approximately 15 air changes are made every hour with a complete change of outside air every 15 min. A slight pressure is maintained at all times so that the excess air flows to the kitchens which are maintained at a slight vacuum. All of the ducts are insulated.

Two large fans of 20,000 c.f.m. capacity and two exhaust fans are provided to supply ventilation air in case the refrigerating machinery has to be shut down.

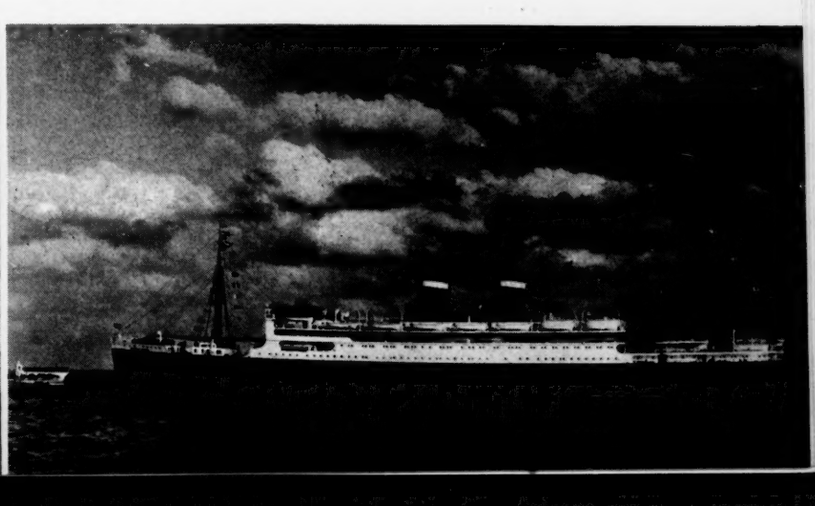
An automatic control system operated by compressed air regulates the air conditions aboard the ship. A dewpoint thermostat placed at the outlet of the washer regulates the saturation temperature of the air by operating the outside and return air dampers and the steam or cold water supply valves.

Some other well known ships which have air conditioning are the United States Lines' *Manhattan* and the Matson Liner *Mariposa.* The *Manhattan,* which is in operation between New York, England, France and Germany, has two of its dining rooms conditioned. Each room has two air conditioning units serving it. The *Mariposa,* operated between San Francisco, Hawaii and Australia, has an air cooled dining room seating 475 passengers. The room is 116 ft. long, 69 ft. wide, and 8¾ ft. high. Carbon dioxide is the refrigerant.

The Colombia



The California



Practical Suggestions on House Insulation†

By J. D. HOFFMAN‡

IN ordinary residences there are three locations where insulation may properly be applied: the wall; the attic floor-ceiling; the roof. Under what conditions may each of these locations apply?

The Wall

If the walls are *solid masonry* the insulation will be on the room side, preferably set off from the wall ½ in. or more on vertical strips, as in furring. These spaces minimize dampness as well as heat transference. Semi-pressed fiber boards, cork board, insulating wall board, and the like, are well adapted to this form of insulation.

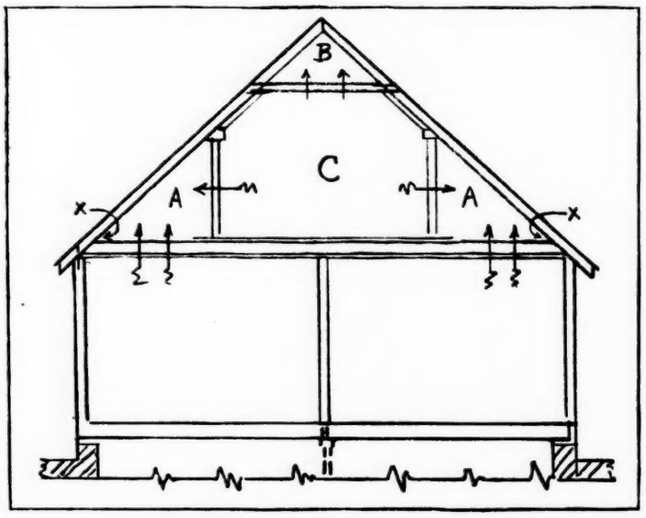
If the walls are *hollow masonry* the interior space, or spaces, within the walls may assist somewhat as insulation, or, what is better, these spaces may be filled with insulating material, preferably of loose fiber or granular form. In any case the inside or room surface insulation should not be omitted.

If the walls are *masonry veneer* (say brick) *on frame*, the insulation may be a fiber board or equivalent between the brick and the frame. Occasionally this board is put on as a substitute for the sheathing. This procedure can be recommended from the standpoint of insulation but by some it is believed to reduce the resistance of the wall to wind pressures, especially in structures of two stories or more in height. In a masonry veneer structure on frame the bonding between the masonry and frame can not be made as resistant to wind pressures as a solid masonry wall, consequently the inner core (frame wall) should be made especially strong. Insulations on the inner surface or between the studs, or both, as given in the next paragraph are preferable.

If the walls are ordinary *framed construction*, insulation may be applied: (1) to the outside of the studs, either as a substitute for the sheathing (with the same objection as given on masonry veneer walls) or in addition to the sheathing. Aluminum foil may be used at this point very satisfactorily in addition to the sheathing, (2) to the inside of the studs as a substitute for the lath, or (3) within the stud spaces. No. (1) is fair and sometimes used, No. (2) is very satisfactory in that fiber board or equal may replace the lath as a plaster base and a good plaster surface may be put on at small additional expense above that of the lath and plaster. A paint or varnish surface may be preferred instead of the plaster. The insulating qualities in (2) are very much improved over that of the lath and plaster and, in addition, the wall is strengthened. No. (3) is probably more nearly ideal than any. Quilted fibrous materials or aluminum foil may be nailed between the studs, the former reducing conducted heat, the latter reducing radiant heat. Any of these will eliminate much of the heat that would pass from the inside to the outside of the wall but none will cut off the convection of heat to the attic if there are any openings between the stud spaces and the attic, which very frequently is the case. Rock wool materials or equivalent filled into the stud spaces make splendid insulation providing the filler does not settle and leave the upper part of the wall uninsulated. A much preferred insulation for this space is rock wool materials in bat form, prepared about 3⅝ x 15 x 18 in. and pressed in place between the studs. This insulation is easily installed, fills the entire space, cuts off convection of the air, will not settle, and probably gives the most insulation value for the money.

It is evident that walls in framed houses already built can be insulated only by having materials in a fibrous or finely divided state blown into the stud spaces. This can be done by removing lap sliding and sheathing boards around the top of the wall and below windows, but can not be done satisfactorily if walls are firestopped.

New types of walls built up on steel studs or frames of various shapes with metal or synthetic stone exterior, and lath and plaster, prestwood or other interior finish, are developing rapidly. These walls are thinner than the standard framed wall and are usually designed with insulation possibilities. For the medium-to-small houses of the future these types may play a very important part. Insulation for such houses is a question of the near future.



Attic Floor-Ceiling

The kind and amount of insulation put on the attic floor-ceiling will depend upon what the attic is to be used for. There seems to be no logical reason for thoroughly insulating both the floor-ceiling and the roof. If it is to be used for other than storage purposes, i.e., for living purposes, the emphasis should be laid

†Abstracted from a paper presented at the 23rd annual convention of the National Warm Air Heating and Air Conditioning Association, January 30, 1936.

‡Professor of Practical Mechanics, Purdue University.

on roof insulation. If it will not be used or if it is to be used for storage only, the emphasis would be laid on floor-ceiling insulation. In existing houses, having the attic partially or wholly floored, the insulating material may be blown in by compressed air, as in walls.

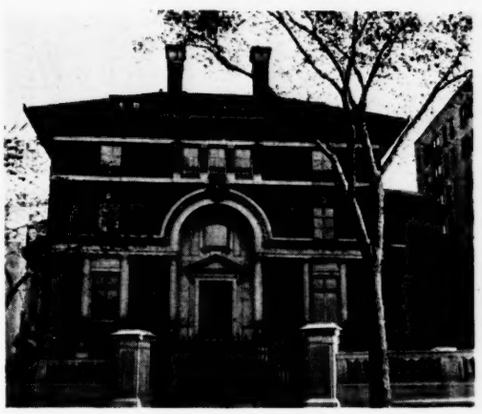
One of the most difficult cases to handle is the bungalow type house with a room or two in the central part of the second floor and attic space on the sides and above the rooms as shown in the sketch. Spaces A and B should be insulated. If the roof is thoroughly insulated, giving special attention at x x, where the roof meets the wall, this will satisfy. In the absence of this, however, floor-ceilings and walls at A and B should be insulated. One common defect in such houses is the neglect of attic insulation with the result that living conditions in winter are very bad. If the floor-ceiling and wall at A and B are properly made, with especial attention at x x, the insulation would be considered proper. Care should be exercised that the heat from the first floor room, which enters the spaces between the joists below room C, does not flow out into spaces A A. Spaces between the joists should be stopped at the room C wall line. A very good insulation for a bungalow house would be 3 to 4 in. of rock wool across the entire ceiling of the first floor and above room C in space B, with wall insulation on the attic side of room C and stops between the joists as stated.

Roof

Roof insulation has the advantage of providing a cooler attic and greater possibilities in keeping the house cool in the summer time. In many cases the first cost of this insulation is greater than that for the floor-ceiling because of the greater area to be insulated. In the case of the bungalow type house, however, it is questionable whether or not first class roof insulation may not be provided at less expense than the insulation of spaces A and B on both floor, wall and ceiling.

1,000 Ft. of Copper Tubing Simplify Heating Modernization Job

AN excellent example of the adaptability of copper tubing in modernizing work appears in the contract just completed in a 30-room private residence at 89 St. and Riverside Drive, New York, by Worcester & Hamilton, contractors, of Astoria, L. I. A decade ago this dwelling of brick and stone stood out as one of the city's imposing structures of its class and even today has few competitors as to size and beauty, standing as it does high above the shores of the Hudson River.



As is well known to heating men, such buildings, due to exposure, size, and other factors are not easy to heat. The owners managed fairly well for many years with an indirect system. Indirect systems, while commonly used for heating first floors, rarely are seen in this locality for heating upper floors. Nevertheless, the Riverside Drive building here discussed was warmed in all rooms via the indirect plan. As the equipment began to show its age, heating results slumped badly and occupants complained of discomfort.

Worcester & Hamilton, in suggesting modernization, recommended a vapor type arrangement for second and third floors, supplemented by an indirect system for the first floor. In making the change the old radiators were removed, the oil fired boiler overhauled, and piping redesigned for vapor.

Some 50 new convector type radiators were installed in register openings, all being recessed. First floor indirect radiators were replaced and old grilles used as much as possible. Little difficulty in carrying forward the work was experienced until the matter of running new risers arose.

Had the indirect stacks been straight, pipe risers could, of course, have been inserted in ducts fairly easily. But in nearly all cases such ducts had bends and offsets in them and as the owners did not wish to have the walls cut, straight supply and return risers could not be inserted. Copper tubing was the obvious answer to the problem.

Altogether it required approximately 1,000 ft. of copper tubing to connect all radiators; pipe sizes ranging from ½ in. to 1¼ in. Sections were joined with flange fittings. Most of the risers were 1¼ in., introduced into ducts at the top and pulled down to the basement. Thus offsets and bends proved no real obstacles and installation difficulties were minor in importance.

Another interesting feature of the modernization was that all radiators are equipped with individual thermostatic controls, the oil burner also being automatically controlled by a wall thermostat placed in a front staircase between first and second floors.

Automatic Controls for Central Fan Systems for Summer Operation

In previous articles[1] on automatic control it has been pointed out that so far as central fan systems are concerned the automatic control actually used results from a judicious selection from among a relatively few kinds of equipment which was classified and defined. It was also noted that the more complex control problems for such systems can usually be solved by combining certain simple combinations of control elements to meet the particular requirements encountered.

Also it is believed that it is easier to understand what appears to be a complicated problem by proceeding from the simple toward the complex and from the better known toward the less well known. With this thought in mind last month's article dealt with some simple control combinations for use with central fan systems intended for winter operation only, since such installations are probably better known than those intended for use in summer.

Now we are ready similarly to consider automatic controls for central fan systems for summer use only. Later, attention will be directed toward these systems when intended for year 'round operation.

In limited space it is evidently not possible to aim at completeness. Consequently, the intention is to illustrate some simple control arrangements and then to suggest, how some of these may be grouped to secure combinations approaching those used in operating installations. Diagrams will be used for illustrating the various control arrangements selected.

Form of the Diagrams

The diagrams are exactly the same in form and outline as those used for illustrating the controls with winter systems in the preceding article. None are to scale nor do they show actual equipment. What they do show is the position and the arrangement of controls relative to each other and relative to the fan, outside air intake, and the return air duct which is shown in all of the diagrams. The fact that a return air duct is included in the diagram illustrates how fully diagrammatic the illustrations are. The duct implies that there must be *some* means of exhaust so long as outside air is being introduced. In some summer cooling systems where there is no artificial cooling capacity, outside air is depended on entirely to accomplish any cooling expected. In such cases large quantities of outside air are circulated—up to the full capacity of the fan—and an equal quantity is exhausted even though there may be no actual exhaust duct. At the other extreme is the case where but little outside air is brought in, as occurs when recirculation of the return air is practiced in order to conserve artificial cooling capacity on hot days. In such cases there is nearly always a return air duct actually present. A draw-through arrangement at the fan is shown in all diagrams for the sake of uniformity even though a blow-through arrangement may be used in practice. As only the summer functions are being considered here, none of the winter equipment is shown on the diagrams even though it may be present.

[1]See HEATING & VENTILATING, February, 1936, page 21, and March, 1936, page 51.

Kinds of Summer Plants

Central fan systems for summer use can be classified in a fashion essentially similar to that used for winter installations. The essential difference between summer installations and winter installations is that the one encounters the problem of reducing temperature and heat content of a mass of air, while the other must meet the problem of raising the temperature and increasing the heat content. Both encounter the problem of supplying suitable air quantities and of distributing it in a satisfactory way.

Summer plants can be grouped conveniently as follows:

1. Plants which do not return used air to the fan, in which the air is obtained from outside, cooled in one way or another, distributed for use, and then all exhausted back to the outside, and

2. Plants which do return used air to the fan, in which the air is handled as in class 1 but a part or all of the used air is diverted to the fan entrance instead of being exhausted to the outside. There are two variations of this kind of plant, viz:

   (a) those using varying proportions of outside air as desired, and
   (b) those using fixed proportions of outside and return air.

It should be understood that class 1 plants have no other source of ventilation air than that brought in from the outside and that an equal volume is returned to the outside. Such plants are essentially ventilating plants to which a great or small degree of air cooling may be added. They are never strictly cooling plants in the sense that they do not ventilate at the same time that they cool.

Plans of the 2(a) type may be so operated that the proportion of outside air drawn in is reduced to zero. At such times they would be strictly cooling plants in the sense that no ventilation air would be passing through them. Such operation is sometimes practiced during periods when few or no people are present in a building but under most circumstances complete recirculation of used air is not considered practicable. Therefore, nearly all summer central fan plants are always concerned with both ventilation and air cooling

Methods of Cooling Air

In central fan systems there are two methods of extracting heat or moisture or both from the moving stream of air entering the fan, viz.:

1. *Water spray cooling,* which is accomplished by passing the air through a mist of finely divided water particles kept at a temperature lower than the air temperature, and

2. *Surface cooling,* accomplished by passing the air across and over a metal surface (usually finned) kept at a temperature lower than the air temperature. Depending on the substance used to keep the metal coil surface cold there are two kinds of cooling surfaces:

(a) direct expansion cooling coils, where a refrigerant is evaporated inside the coil and which method naturally can be used only where there is artificial refrigeration connected to the fan cooling system, and

(b) water cooling coils, generally called simply cooling coils, where the water may come from any available source, such as from wells, from melting ice, or from an "indirect" arrangement for artificially cooling it in connection with refrigeration equipment.

With these two general methods various combinations are possible and it is often found desirable in particular cases to use both methods of cooling the air.

Operating Characteristics

With water spray cooling it is generally necessary to supply a constant quantity of water to any one bank of spray heads as otherwise the heads will not create the desired fine mist and also as all the air mass may not come in contact with cool water particles. It is also characteristic of spray cooling that in a well designed spraying equipment the whole mass of air passed through the water mist emerges at a temperature very close to the temperature of the spray water. Both these characteristics of the spray method have to be taken into account in devising automatic control methods to fit this type of air cooling.

With surface coils the quantity of liquid passing through the coils can be varied by adjusting its rate of flow and the temperature of the liquid can also be adjusted. With such coils, too, the leaving temperature of the whole mass of air passed across them may or may not closely approach the temperature of the substance inside the coil. How closely it approaches depends to considerable extent on the design of the coil, including the height and spacing of the fins, and on how long the air is permitted to stay in contact with the cold coil surfaces.

Staging of the cooling action may be practiced with either method of cooling for spray banks can be placed in series and so also several sets of cooling coils instead of one can be installed where desirable to do so. Since staging of the compressor is also possible and since water at various temperatures is obtainable by mixing, or otherwise, it is evident that much can be done with this feature when it is desirable to practice it.

Humidity and Sensible Heat

Since control of humidity and removal of definite quantities of sensible heat and of latent heat (humidity) are so often an essential part of a summer central fan system, the effects of the cooling methods on these items has to be understood.

With any of the cooling methods just outlined some removal of sensible heat occurs with any lowering of the dry bulb temperature. Moisture, or latent heat, however, is removed from any particle of air only if that particular particle has its temperature brought down to the dewpoint. If the temperature of the whole mass of air is brought down to the dewpoint, moisture will be removed from all the air mass.

In the case of spray cooling where the whole mass of air comes in contact with the cold water, moisture is removed from virtually all the air particles if the temperature of the spray water is kept sufficiently low. In surface cooling moisture may be removed from only a part of all the air particles which pass the cold coil. Because of this difference in the effect on relative removal of moisture and sensible heat quantities when spray cooling or surface coil cooling is practiced, it is necessary to devise different control methods to suit the kind of cooling being used and the relative quantities of sensible heat and moisture which are removed.

Methods of Securing Operating Control

Methods are by no means standardized as to detail but in general they are classifiable by the cooling method used. The following list is comprehensive enough to cover most of the principles in use.

1. With surface water coils, operating control is secured by (a) adjusting the water quantity (by valves), (b) the water temperature (by mixing), or (c) by operation of the refrigeration compressor or the ice meltage rate, while humidity control is linked with this where necessary by (a) coil staging, by (b) varying the rate of air delivery over the coils, and by (c) the design and arrangement of the coils.

2. With direct expansion coils, operating control is obtained by essentially the same methods except insofar as methods are limited by the refrigeration equipment itself.

3. With spray cooling, operating control is secured by adjusting the spray water temperature, usually by mixing finally, although auxiliary control of the sources of warm and cold water are necessary and this may call for control of the refrigeration compressor, and other auxiliaries.

As in the case of heating installations the problem is always to select from the equipment available that which will accomplish the desired results with economy of first cost and of operating cost. While this discussion of cooling methods and their control is by no means complete, enough has been given to make it apparent that within these principles there is latitude for many solutions to the control problem in summer plants. Operating plants have made use of a great variety of these possibilities. In limited space it is amply evident that diagrams cannot possibly be included to show either all the combinations of controls

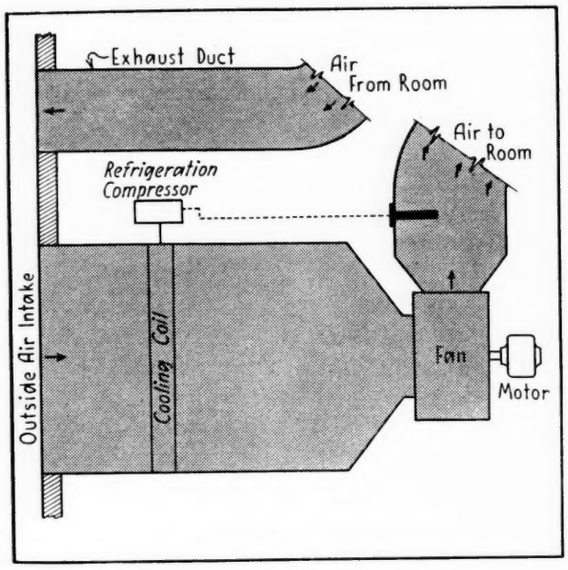


Fig. 1

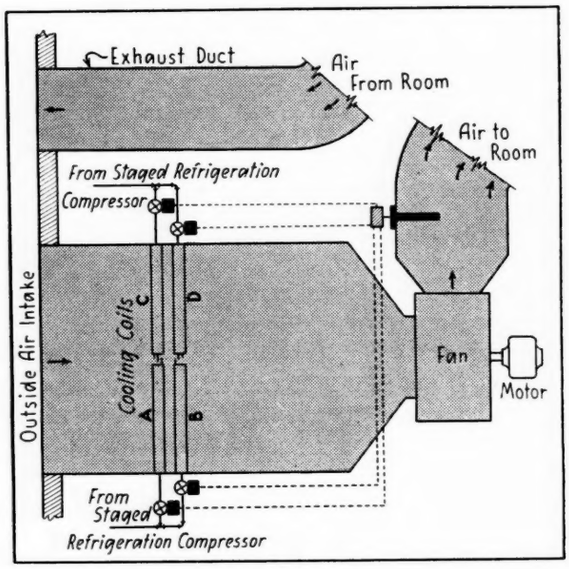


Fig. 2

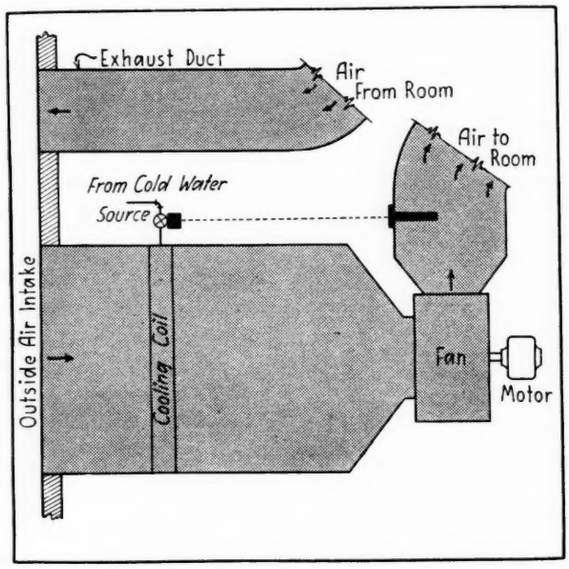


Fig. 3

possible or even all those in common use. It is possible though to illustrate direct expansion cooling, cooling coils, and spray cooling but none of the many possible combinations. Neither is it possible to give much attention directly to humidity control. One of the diagrams does illustrate an application of recirculation of a part of the return air.

As a result of the necessity of including but a small number of diagrams, the attempt has been made to make those shown cover as wide a range as possible while still retaining the essential idea of avoiding complications in the diagrams and of proceeding from the simple toward the complex.

Fig. 1 illustrates an extremely simple control combination. Here a temperature controller in the fan discharge duct (or in the cooled room) operates to start and stop a refrigeration compressor which supplies refrigerant to a direct expansion surface coil extending across the whole cross section of the intake air duct. Obviously, the temperature controller can be of the two-position or on-and-off type. Humidity controls and temperature limiting controls can be added.

Evidently an alternate method of control would be to let the compressor start and stop under the action of a pressure controller set to maintain a closely limited control of the back pressure on the compressor and to let the temperature controller, shown in Fig. 1, be connected to a valve in the pipe leading to the cooling coil. In such an arrangement the temperature controller should be of the modulating type in order to position the valve in intermediate positions between wide open and tight shut. Additional refinement in control is possible by using a multi-stage compressor or by using more than one compressor.

Fig. 2 illustrates a case where the direct expansion coil is staged and is broken up into four separate coils individually controlled by valves operated by a temperature controller which can be located either in the fan discharge as shown or in the cooled room. It is possible so to arrange control equipment in this plan that the individual valves can be made to open in almost any desired sequence. Also since the pressure of the refrigerant admitted through each valve can be different, the surface temperature of each coil can be different from that of any other coil. This arrangement of coils permits considerable choice in the results obtained. This same general arrangement can evidently be used when there is any source of cold and warm water. In other words it is not confined to use with direct expansion coils but may also be practiced with water cooling coils. Moreover, if the coils on one side only of the duct are cooled that part of the air passing the idle coils can be used to reheat the chilled air, having an effect like the bypass shown in Fig. 6.

Fig. 3 will be noted as being very like Fig. 1 with a water cooling coil instead of a direct expansion coil. Here the water may be obtained from any suitable source and enters the coil under the command of a controller located either in the fan discharge or in the cooled room. Staging may be applied as may also limiting controls for temperature and humidity.

Fig. 4 includes the arrangement of water cooling coil and inlet control valve just shown in Fig. 3 but with

the controller inserted in the return air duct instead of in the fan discharge. Use of return air is not essential to the principle illustrated, however, for the controller could also be located in the cooled room, if desired. In addition there is a temperature controller shown located in the outside air intake. This controller is exposed to the outside air temperature and can be arranged to act as a master control regulating the temperature at which the controller in the return air duct positions the valve leading to the cooling coil. The net effect is that the outside air temperature determines the indoor temperature maintained. Such an arrangement permits varying the indoor temperature as the outdoor temperature varies without maintaining a fixed difference between them. This arrangement of controls is variously called "differential" and "compensated" and the principle may be applied where spray cooling is practiced as well as with surface coils.

Fig. 5 illustrates a control combination where a temperature controller located in the stream of entering air positions a damper motor in order to control the relative quantities of outdoor and return air cooled. It is adjusted to admit increasing amounts of outside air as the outside temperature falls. The controller in the return air duct (or in the cooled space) regulates the operation of the staged refrigeration compressor to maintain a fixed temperature in the cooled room while the one in the fan discharge is a limit control. Humidity control in the form of humidistats can be readily fitted to this arrangement as desired but is not shown in the diagram in order to preserve simplicity. It is apparent that this same combination of controls can be applied to water cooling coils by the use of suitable valves provided mixing to obtain the desired water temperature can be practiced.

Fig. 6 shows an arrangement where both return air and recirculated air for reheating are used. A controller in the entering air stream positions the damper and thereby regulates the amount of outside air admitted. It is adjusted to admit increasing amounts of outdoor air as the outdoor temperature falls. A fixed temperature is maintained at the discharge from the air washer by the controller which adjusts the relative quantities of warm and cold water pumped to the sprays. The controller in the exhaust duct positions the damper motor to regulate the relative amounts of return air and recirculated air used and thus to keep a fixed temperature in the cooled room. The controller in the fan discharge acts as a low limit to prevent the discharge air falling below a fixed minimum point.

It will be noted that this diagram illustrates spray cooling and that the whole mass of air passing through the washer approaches the temperature of the spray water. To remove moisture this temperature is set at or below the dewpoint temperature of the air. Adjusting the operating temperature of the controller located at the exit to the washer permits control of the moisture content of all the air passing through the washer. As this temperature is lower than wanted to discharge the cooled air into the room with ordinary distribution methods, reheating is obtained by using the relatively warm air introduced through the recirculating duct.

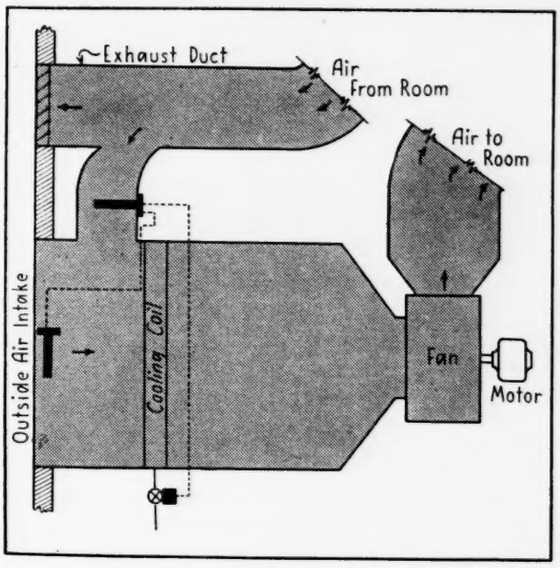


Fig. 4

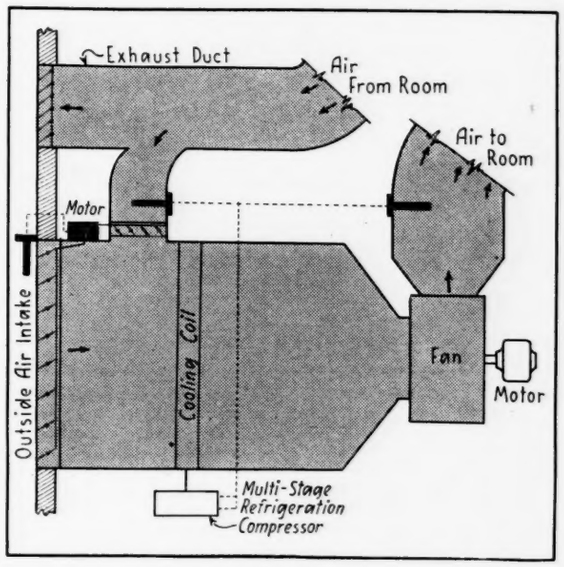


Fig. 5

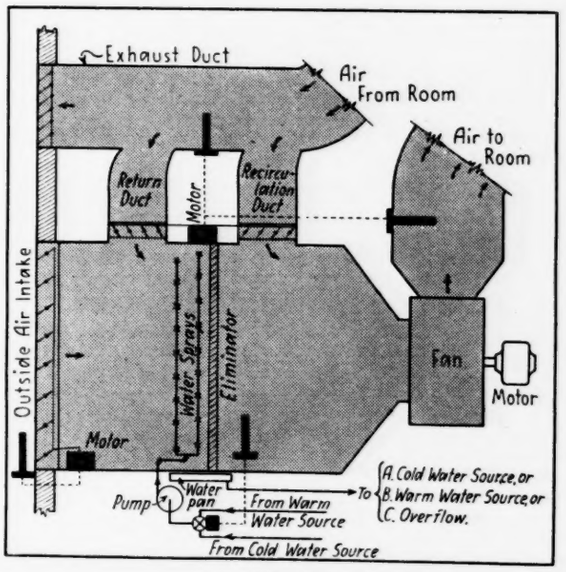


Fig. 6

1936 Business Off to Good Start as Building Contracts and Equipment Sales Rise

THE year 1936 holds considerable promise for a prosperous year if activity in the industry during the first month of the year is any indication. The promises are given support for February and March by the records of new building contracts in February, which should be followed by a corresponding amount of business in the heating industry. Equipment sales for January for some of the major items are summarized in the accompanying table, as are the new building contracts for February.

Unfortunately, complete data are not as yet available for January for all items of equipment, so that HEATING & VENTILATING's Index of Heating Business Activity cannot be computed this month. It is expected that this will be ready for the coming issue. It will be necessary to revise this index since the oil burner figures have been sharply reduced, as it was found that an important number of distillate burners were being counted in with the burners used in computing the heating index. The revised oil burner figures have eliminated this error, and will bring the index for the months of 1935 down to some extent.

On the other hand, statistically minded readers will be pleased to see that for the first time monthly reports are available on both unit and central type air conditioners. These are presented in the table below. They are available only for January, 1936, but will be published on these pages from time to time. No data are available on these series for months prior to January.

Fuel Burning Devices

A 70% increase in oil burner sales for January as compared with a year ago is reported. The boiler-burner units increased 51%, while furnace-burner units showed a gain of 388%, although it should be pointed out that the sale of the furnace units is numerically less than boiler-burner units.

Residential stokers of the type burning less than 100 lb. per hr. of coal increased from 1,241 in January, 1935, to 2,065 in January, 1936, a rise of over 66%. Class 2 stokers (those burning from 100 to 200 lb. per hr. of coal) increased 27%, while the large commercial stokers showed a gain of 38%. No data are available on the sale of gas burning appliances, as manufacturers of this equipment do not make these figures public.

In January, 1935, sales of 190 steel boilers totalling 63,385 sq. ft. of heating surface were reported, while

(Concluded on page 76)

HEATING BUSINESS TRENDS, JANUARY AND FEBRUARY, 1936

EQUIPMENT	JANUARY, 1936	JANUARY, 1935
Ventilating blowers, orders in dollars	86,003	60,571
Small housed blowers, orders in dollars	60,707	42,070
Propeller fans, orders in dollars	54,858	50,670
Unit heaters, orders in dollars	399,998	284,005
Unit ventilators, orders in dollars	202,503	158,417
Indirect heating surface, orders in dollars	108,925	84,990
Self-contained air conditioning units, orders in dollars	36,263	*
Other air conditioning units, orders in dollars	215,420	*
Industrial central air conditioning systems, dollars	78,706	*
Comfort central air conditioning systems, dollars	307,941	*
Refrigeration otherwise sold for air conditioning, dollars	172,445	*
Air washers, orders in dollars	39,357	*
Continuous air filters, orders in dollars	927	*
Unit air filters, orders in dollars	16,707	*
Humidifiers, orders in dollars	83,192	*
Oil burners, number of units shipped in U. S., mechanical draft	5,248	3,089
Boiler-burner units, number shipped in U. S.	411	272
Residential stokers, sales in number of units	2,065	1,241
Steel boilers, orders in square feet of heating surface	214,719	63,385
Cast iron round boilers, shipments in thousands of pounds	3,120	2,666
Cast iron square boilers, shipments in thousands of pounds	13,786	10,604
Cast iron radiators, shipments in thousands of square feet	4,183	3,117
Non-ferrous convectors, sales in square feet, elements only	145,028	80,595
Non-ferrous convectors, sales in square feet with cabinets	136,021	93,407
Cast iron convectors, sales in square feet	292,724	111,732
NEW BUILDING CONTRACTS (MILLIONS OF DOLLARS)	**FEBRUARY, 1936**	**FEBRUARY, 1935**
Commercial	12.6	9.2
Industrial	13.4	7.7
Educational	21.1	5.7
Institutional	5.7	0.6
Public	4.3	4.8
Religious, Memorial	0.8	0.5
Social, Recreational	4.7	1.9
TOTAL, Non-Residential	62.6	30.6
Residential	31.2	16.6
TOTAL, All Building	93.8	47.2

*Data not available.

EDITORIALS

Short Courses

Short courses at colleges and universities this spring reflect the growing demand for instruction in heating, ventilating, and air conditioning. Iowa State College just closed a three-day course held March 25-27; Michigan State College held one from March 31-April 3; University of Illinois plans one for May 4-5. More and more men take advantage of these courses each year while home study and evening courses also continue to report increases in numbers of students. Manufacturers are now at some pains to instruct their own dealers in proper methods of applying their equipment. All in all these various courses are doing a great deal to supply the deficiency in trained personnel to handle the new application problems.

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Odor Control

It has been said that one major idea is about all that any one generation is ever able to grasp. Perhaps this thought has something to do with the length of time that has been required for general acceptance of the idea that odor control or elimination is primarily what fixes the amount of ventilation air. At least it fixes the amount of this air when human ventilation is practiced by the methods and with the equipment now in common use. It is fully a generation since odor control began to be studied and since practicing engineers and contractors started to see what could be done about it. Its importance as a limiting factor came to be understood but slowly, though. Now the idea is fairly generally accepted and there is a general grasp of how odor fits into the schemes of accomplishing human ventilation.

Acceptance of the idea has been accompanied by changes in method as both better means and better understanding have developed. Now the problems of odor control are being attacked on several fronts what with more interest in the problem and with more general confidence that the right evaluation of odor in ventilation has been achieved. Odor filters, odor masks, and outside air are all being tried and retried. Slowly but surely information is accumulating which permits better application engineering. Now that the elements of the problem are being clarified we can look for a much more rapid rate of increase in the useful data pertinent to the subject.

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Fundamentals

Having recently reviewed most of the fundamental data on which application engineering in our field rests, we were especially interested when a friend not directly engaged in this kind of engineering remarked that it is a shame that the theory has not been more fully analyzed.

Evidently he understands rather well that much of our information, in spite of its being now fairly well systematized, is still essentially empirical. Whether or not he fully understands how large an element of individual judgment is still necessary in applying it is a point on which we have no knowledge.

We doubt if he does fully understand it, for we believe that relatively few of those who work with this material day by day and who get used to doing certain routine jobs with it, really realize how little of sound reasoning there is back of it all. What they do know is that they can make some layouts based on the information in such a way that the resulting installation works reasonably well. To the great majority of users there seems to be little practical point in knowing anything more than this for the routine job can be reasonably well handled without too great a tax on judgment. The rub comes where the job is not routine, or where there is a call for cutting the estimate to the bone, or where there must be a specific operating guarantee. Application of the data to such situations is when the real lack becomes apparent at once.

We are not inclined to agree with the idea that the lack of better grounded material is a shame, or that a better analysis of the fundamentals is going to produce better data at once. But we do know perfectly well that a good understanding is essential to good use of data and that a good analysis of principles helps mightily in straight thinking.

In our judgment improvements in our fundamental data are no more to be desired than are improvements in the methods of making use of what we now have. We believe that both improvements can and should move along together. The borderline should be carefully examined in any way available and not merely by analysis. Properly used, the results of our present applications can be made exceedingly useful—probably as useful as any resort to pure analysis alone. What is needed is a good coordination of the work of the pure analyst and of the man who proceeds by present rule as best he can.

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The Marketing Problem

One of the acute problems which is facing the industry as a whole and which has been particularly pressing to sales managers for some time is the riddle of how to get air conditioning equipment on the market economically but effectively. The problem has many phases but one of the most troublesome has been the question of how to arrange for suitable personnel to handle application engineering on the job and to secure suitable installation and service after the operation starts. These phases are not selling problems but are closely related to selling, and are more properly thought of as parts of the general marketing problem. As such they are tied up with all phases of the problem, and in many cases it is not possible to isolate the individual items for separate solution because of the nature of the product, company policy, relation with the marketing of other items, and the necessity of meeting trade customs and practices.

Really the marketing problem would seem to be an individual one for each maker of equipment to solve in accordance with his own necessities. Now there is not only indication that the truth of this statement is steadily coming to be better appreciated but that many are convinced that marketing methods must be kept flexible enough to cover all kinds of local variations from section to section of the country. Certain it is that methods found to work in one locality do not lend themselves at all well to conditions elsewhere.

Another point about the marketing of air conditioning equipment which should be understood is that there is every likelihood that the methods used will continue to shift for some time to come. The depth of the immediate market is not yet known with any certainty and to a considerable extent methods have to change with the nature of the buildings being equipped. A personnel well suited and well trained to handle applications in residences and to keep them running may not be at all suited to doing the same things in other kinds of buildings.

It looks as if the engineering, installation, and service phases of the marketing problem in air conditioning are not only individual but that they are likely to continue to present problems for a considerable time. Any generalized solution seems rather far off.

ABSTRACTS of Current Papers, Books and Pamphlets

Handbook of Applied Mathematics

The purpose of this book is to render the same service in trade and business calculations that the engineering handbook does in engineering computations. It offers formulas and short cuts for solving all types of everyday problems encountered in a wide variety of fields.

Formerly this book appeared in two volumes. In this, the second edition, all of the old, plus a considerable amount of new material, has been put into one volume.

Some of the many subjects covered are mathematics—algebra, geometry, trigonometry, weights and measures, concrete, carpentry, plastering, painting, plumbing, heating sheet metal work, electricity, business mathematics, and accounting.

[*"Handbook of Applied Mathematics," by Martin E. Jansson. Second edition, published by D. Van Nostrand Co., Inc., 250 Fourth Ave., New York. Flexible binding; 5 x 8 in.; 1,010 pages; price, $6.*]

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Household Gas Water Heating

While this book was written primarily for salesmen of household gas water heating appliances, it should prove of interest to all persons dealing with the design and installation of domestic hot water systems. This is true because the author has included in the book a large amount of data and factual information on water heating. His main idea in presenting these data is to give the salesman a knowledge of the engineering aspects of the subject so that he will understand the performance and use of his product. However, he has done such an excellent job in writing up this information that he was able to cover the subject rather completely and thus make the book of value to more advanced workers.

Perhaps the best way to show how completely the subject is covered is to list the chapter headings. Engineering ABC's; water heater types; hot water—how much?; gas consumption for water heating; water piping and installation; what size heater; automatic heater accessories; competitive water heating; and hot water—what for?

[*"Household Gas Water Heating," by Charles W. Merriam, Jr. Published by American Gas Journal, 53 Park Place, New York. Cloth bound; 5 x 8 in.; 142 pages; price, $1.50.*]

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Heating in Racine by Income Classes

The eighth of the series of consumer uses by income classes has been released by the Department of Commerce, and covers the city of Racine, Wis. In this city, tenants and homeowners are alike in providing evidence that, irrespective of materials used in house construction in this city, the central heating plant is the most commonly used equipment for house-heating. Approximately 85% of these homes have central plants. As compared with other types of heating apparatus, by far the highest percentage of these homes have warm air furnaces. Next in importance among central plants, on the basis of present use, are hot water plants. Nearly 4% of the families use steam or vapor systems. Approximately 15% of the Racine families rely on heating stoves.

More than three-fifths of the families that use stoves are tenants, and over 85% of the stove-using families reported incomes of less than $1,000 for the year covered.

The relationship between family income and the use of central heating apparatus in Racine homes is indicated by the fact that heating plants are used by more than 98% of all families with incomes of more than $2,000. Moreover, throughout successively lower income ranges below $2,000 there is a continuous drop in the proportion of homes supplied with central heat. All families in the $5,000-$6,999 income class have heating plants.

[*"Consumer Use of Selected Goods and Services, by Income Classes, Racine, Wisconsin," by Ada Lillian Bush, Chief, Consumer Market Section, Marketing Research and Service Division. Available from the Bureau of Foreign and Domestic Commerce, Washington. Price, 10 cents; standard size; 32 pages; multigraphed.*]

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Fuel Oil—Storage and Use

This small book should be in the hands of every person interested in the installation of oil burning equipment, for it is one of the best sources of information on safe practices in oil burner installations. It was prepared by a group of fire insurance companies with the idea of indicating good practice for the installation of such equipment.

The recommended practice outlined is in general accord with local and national regulation, with some modifications to further reduce fire hazards.

The booklet is divided into six main sections: characteristics of fuel oils; fuel oil storage; oil distribution; oil burners; fuel oil appliances and safety devices for oil burners. Each section is further subdivided and the individual subjects fully discussed. A large number of illustrations and tables are used to illustrate and supplement the text.

[*"Fuel Oil—Industrial Storage and Use," prepared by the Inspection Department, Associated Factory Mutual Fire Insurance Companies, 148 High St., Boston, Mass. Heavy paper cover; 4¾ x 7¼ in.; 82 pages; price, 35c.*]

•

Increasing Clinker Formation in Small Bituminous Stokers

Small underfeed bituminous stokers, such as are used in residential buildings and small industrial plants, are designed so that the ash is removed in the form of clinkers. The ash moves out of the retort of the stoker onto the refractory or cast iron hearth or dead plates where it fuses into solid clinkers which can be readily removed with a pair of tongs.

When this type of stoker is installed in residential buildings, there is the possibility that only a small per cent of the ash will be fused into clinkers during periods of mild weather when the stoker operates only a few minutes every hour. Under these conditions, when combustion is proceeding at a very low rate, the bed may tend to become thinner and allow more air to flow through it and, consequently, lower the fuel bed temperature. The fuel bed temperature may even fall below the ash fusing temperature so that no clinkers can be formed. As the fuel bed temperature approaches this point, the accumulation of unfused ash may smother the fire.

There are three methods of preventing this occurrence. The first is to restrict the air to a greater degree than usual, to obtain a heavier fuel bed and higher temperature. The second is to operate the stoker at a higher rate of feed. The disadvantage of this method is that it tends to overheat the building. If this occurs, the hold-fire control should be set for a shorter period of operation. A responsive limit control such as a thermostat in the smokepipe of the boiler or furnace will serve to prevent overshooting of the temperature. The third method calls for the addition of some foreign material to the coal which will cause the coal

(Continued on page 76)

WHAT READERS SAY . . .

Kenmore Cooled by Kelvinator

I just received the March edition of HEATING & VENTILATING and had my attention called to your article on the Kenmore Hotel in Boston, Mass.

In reading this I was very much surprised to find reference to Frigidaire equipment as being used for cooling wines and beer, whereas no mention whatsoever is given of Kelvinator equipment, although the article describes a Kelvinator air conditioning installation.

In fact, the phraseology of the reference to the cooling unit for wines and beer would imply that the entire installation was a Frigidaire project.

JOHN S. GARCEAU,
Advertising Manager,
Commercial Divisions.

Kelvinator Corp.,
Detroit, Mich.

Mr. Garceau is correct, the article wrong. Kelvinator equipment was used in the installation described, with Frigidaire equipment used for cooling the wines and beer.

•

Radiator Data

Where can I get a list or table whereby it is possible to rate steam or hot water radiation that is already installed? Is there any reference published that gives the square feet of radiation on most of the existing sizes? I find this information necessary to get total radiation in any given job for figuring stoker sizes necessary. It is more accurate this way than by figuring only for the existing boiler size.

Toledo, Ohio J. W. S.

We are sending you a copy of one of HEATING & VENTILATING'S blue Reference Data sheets which gives the sizes and ratings of practically all the tubular type cast iron radiators now built. We are also sending a table showing the ratings in square feet per section of the old columnar type cast iron radiators put out by one company.

You might also be interested in the "Pape-Swift Radiator Reference Book." This book is priced at $1 per copy plus 10 cents postage, and can be obtained from John S. Swift Co., 230 West 17 St., New York. It is a planographed book of about 300 pages and gives the radiator ratings and dimensions of practically all the cast iron radiators manufactured.

•

Gas Engineering Reprints

Last month the editors replied to a question written by W. A. Malloy, Central Hudson Gas & Electric Company, that reprints of the series of articles by A. G. Canar on "Engineering Gas Conversion Burner Installations" would be obtainable only in back numbers of HEATING & VENTILATING. Requests for copies of these articles continue to come in. Now we are considering reprinting the whole series in a pamphlet, and will do so if there is enough interest shown. If you want a copy of such a pamphlet write to us about it, stating how many you would purchase if available at a reasonable price.—EDITOR

•

Design Temperature

According to your Reference Data Sheet No. 58, the outside design temperature for Terre Haute, Ind., is given as —10° and the lowest recorded temperature is given as —18°, while the design temperature for Indianapolis is given as —5° and the lowest recorded temperature is given as —25°. On Reference Sheet No. 67 the normal degree-days for Indianapolis is 5,298, and for Terre Haute 4,872. This would seem to indicate that normally Terre Haute should have a higher design temperature than Indianapolis.

Are these values correct? If they are correct, would you kindly indicate your method of determining these design temperatures. If they are not correct, what are the correct values? I am anxious to use these design temperatures on a year 'round air conditioning job and would appreciate a prompt reply.

Terre Haute, Ind. A. F. W.

It is not unusual to find that a city which has a lower minimum temperature than another has a higher design temperature. This may be due to the fact that the extreme low temperature was more or less of a "freak" affair in the one city, and is not necessarily an important factor in setting up the design temperature.

It is also to be noted that the degree-days of two cities have nothing whatever to do with their design temperature. A city with an equable climate may pile up degree-days at an even rate most of the year, and have the same number of degree-days as a city which has a short but severe winter. At any rate, this should not be taken into consideration in connection with the design temperature.

The figures for Terre Haute and Indianapolis published on our data sheet were determined by letter surveys. As stated in the article which accompanied this information when it first appeared, we have no way of knowing whether or not this design temperature is really justified. We queried several people in each city, and where there was a disagreement took what seemed to be the prevailing opinion. Terre Haute being farther south than Indianapolis would apparently have a higher design temperature than Indianapolis, which it does, and we believe this figure to be at least fairly near what would be theoretically correct.

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Gas Radiators

Can you furnish me with any figures on the number of gas radiators being sold?

New Orleans, La. V. M.

As reported in the Census of Manufactures for 1931, the value of gas fired radiators produced in that year was $307,775. A corresponding figure for 1933 is not available, according to the Department of Commerce, but from data furnished by the Department it is estimated that the value in 1933 was $275,000. Figures for 1935 are not yet available. From the value you may be able to estimate, at least approximately, the number of these sold.

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Railroad Air Conditioning

How many cars have the railroads air conditioned so far? If possible, I would also like to know how many of these are ice cooled and how many are cooled mechanically.

Woodmere, Long Island F. T. C.

According to one authority, there were in service at the close of 1935, 5,819 conditioned passenger cars as follows:

TYPE	RAILROAD CARS	PULLMAN CARS	TOTAL
Steam ejector	467	312	779
Mechanical compressor system with direct drive	412	1,578	1,990
Compressor system with electric drive	733	416	1,149
Compressor system with internal combustion engine using liquefied petroleum gas	3	—	3
Water ice	867	1,031	1,898
TOTAL	2,482	3,337	5,819

NEWS OF THE MONTH

Heating Is 80% of Air Conditioning, Chicago Group Told As Radiator Heat Campaign Opens

CHICAGO—Details of the campaign to promote the sale of air conditioning with radiator heating were explained at an enthusiastic meeting of one hundred representatives of manufacturers and wholesalers of air conditioning and radiator heating equipment at the Union League Club here March 12.

Speakers explained that the object of the campaign is to enable everyone participating to sell more heating installations. One of the means of accomplishing this objective is to persuade the radiator heat trade to tell a uniform, standardized story on the advantages of radiator heating, on air conditioning with radiator heating, and on heating as the most important factor in air conditioning.

R. D. Berry, president of the Chicago Master Steamfitters Association, as the first speaker of the evening, told how the campaign started. "The campaign which we are now launching in Chicago under national auspices had its beginning here last spring," Mr. Berry said. "At that time a group of contractors, convinced of the superiority of air conditioning with radiator heating, met to discuss ways and means of widening the market. Subsequently, we called a joint meeting of contractors and wholesalers. In attempting to find means of getting the campaign under way, we were confronted with the practical difficulty that a duplication of the campaign in other cities would necessarily involve a heavy duplication of expense for manufacturers doing a national business.

"Under the circumstances, it was decided that the most practical method of accomplishing the objective would be for national interests to underwrite the preparation of all the educational and promotional materials and with local interests merely to finance the practical application of the campaign to each city or district.

"We are now ready to take the materials prepared for us and to launch this campaign in Chicago immediately. By means of these materials, we will all be telling the same story. I am convinced that the program will be a success."

S. Lewis Land, educational director of the Heating, Piping and Air Conditioning Contractors National Association, explained that the educational and promotional material developed by the program committee represents the crystallized thought of the industry. The manner in which all manufacturers opened their files of data and gave the committee free access to everything bearing on the sale of radiator heating equipment is unprecedented in the industry, according to Mr. Land. With the aid of professional assistance, the committee assembled the data into an 80-page manual which will be the Bible of the campaign.

C. G. Newton, vice-president, Stevens-Root Co., emphasized the fact that the campaign marshals all the forces of the industry. He pointed out that individuals have very little influence on the general trend toward or away from radiator heating, and that no individual firm has man power enough to contact the public, the architects, and the builders, to say nothing of being able to finance an advertising campaign. He showed that the campaign, by pooling the efforts of the industry, gives each member firm the resources of the group.

Martin Weil, vice-president, Weil-McLain Co., declared that he has never seen the radiator heating industry take hold of any project in such an enthusiastic manner. In an average residential radiator heating installation, the boiler amounts to about 23% of the total cost of the heating job; radiators, 16%; specialties, 12%; pipe, valves, and fittings, 15%; pipe covering and miscellaneous items, 15%, and labor, 19%, Mr. Weil said in explaining the interest which various groups should have in the success of the campaign.

Speaking as secretary of the Chicago Master Steamfitters Association Louis T. Braun emphasized that the benefits of the campaign are not to be limited to members of this association. He explained that the territory to be covered in the campaign is metropolitan Chicago which is interpreted to mean the district as far north as the Wisconsin line, west to the Fox River Valley, and south to the Indiana boundary. The Chicago Master Steamfitters Association has underwritten half the cost of the campaign in this district.

Kenneth MacKenzie, Chicago branch manager, American Radiator Co., pointed out that there has been a great development of products in the radiator heating and air conditioning industries in the last few months which indicates that radiator heating is in a position to obtain a large volume of business.

Plans for newspaper, magazine, and radio publicity to tie up with the campaign were outlined by Norman J. Radder, secretary of the Plumbing and Heating Industries Bureau.

Russell G. Creviston, sales promotion manager, Crane Co., said that the theme of the campaign is that heating is 80% of air conditioning. Mr. Creviston explained that the National Bureau of Heating and Air Conditioning is to be incorporated under the laws of the State of New York and that many of the leaders in the radiator heating and air conditioning industry are among the incorporators.

Contractors Back Campaign

CHICAGO—Nearly five hundred heating contractors turned out for the first meeting in the campaign to promote sale of air conditioning with radiator heat, held here at the Sherman Hotel, March 24. With R. D. Berry, president of the Chicago Master Steamfitters Association presiding, those who attended heard preliminary plans for the drive.

J. L. De Neille, president of the Heating, Piping and Air Conditioning Contractors National Association, said: "This can easily be a famous night in the history of the industry, and properly this movement is begun in Chicago for here it was born, developed, and matured." He went on to point out the necessity for contractors to talk in a common language and so gain the public's confidence, and said that the meeting was the first of a series from coast to coast.

Following Mr. De Neille, S. Lewis Land, educational director of the HPACCNA, outlined the campaign with slides reproduced from the new handbook to be used in the campaign, "Practical Air Conditioning For The Home." Mr. Land explained that while the book was elementary, it was necessary to present the subject of air conditioning to the public in this manner so that the point would not be lost in needless technicalities.

Other speakers were C. G. Newton, vice-president, Stevens-Root Co., chairman of the Chicago Committee of the National Bureau of Heating & Air Conditioning; L. T. Braun, executive secretary, Chicago Master Steamfitters Association, and Kenneth MacKenzie, Manager, Chicago office, American Radiator Co.

Over four hundred salesmen of manufacturers and wholesalers attended a similar meeting held March 25 at the Sherman hotel to acquaint this group with the campaign plans.

The meeting of manufacturers and wholesalers was the first of a series of meetings planned for the immediate future and marked the official opening of the campaign in the Chicago district. W. J. Spillane, sales manager, Jas. B. Clow & Sons, was chairman of the meeting of wholesalers and manufacturers.

Buffalo A.C. Standards Considered

BUFFALO—The monthly dinner meeting of the Western New York Chapter, ASHVE, was held March 9 with President Voisinet presiding. The meeting was thrown open for discussion of the Chapter committee's "Comfort Standards for Air Conditioning." Many verbal and written comments were made, which were recorded and will be presented at the next air conditioning committee meeting. It was moved by Joseph Davis, seconded and passed unanimously, that the Chapter endorse the committee's accomplishments to date. It is expected that the "Standards" will be ready for final adoption at the April 13 meeting.

E. P. Hannig, district sales manager, Reading Iron Company, presented three reels of motion pictures describing the manufacture of wrought iron pipe from mine to finished product.

A.C. Manufacturers to Meet

WASHINGTON—The Refrigerating Machinery Association will meet May 14-16 at Hot Springs, Va., according to D. Norris Benedict, the association president. At this meeting, much of the discussion will center around the standardization program which the association has undertaken and on which substantial progress is reported during the past year.

The Air Conditioning Manufacturers' Association will hold its annual meeting at Hot Springs simultaneously on May 16, President P. A. McKittrick has announced.

Barometric Pressure Variation Has Close Relationship to Health of Human Beings

CHICAGO—Dr. William F. Petersen, of the department of pathology, College of Medicine of the University of Illinois, addressed the Illinois Chapter, ASHVE, at its monthly meeting, March 9. His subject was "The Effect of Weather on Man."

Doctor Petersen opened his discussion by stating that everything he was to show during the evening was known to the Greeks; that Hypocrates wrote knowingly of the adverse effect of varying temperatures on the digestion and general well-being of the populace. He spoke of the "universal hot winds" and the "local hot winds" as affecting large areas and small areas of the population of Greece, so that this correlation of weather on man is of ancient knowledge.

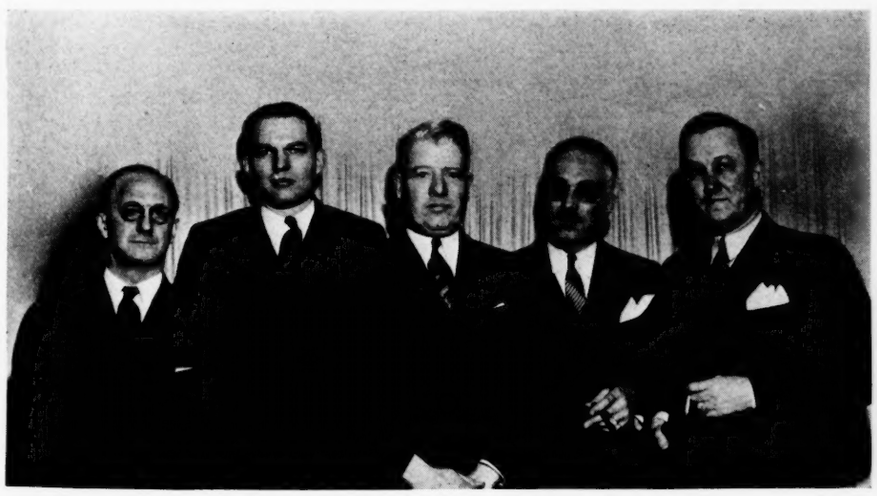
The discourse of Doctor Petersen was profusely illustrated by many lantern slides which clearly showed the many points of correlation that he wished to emphasize. A cyclonic circulation map showing the concentration of cyclonic lines in the northern United States clearly showed the path from the West Coast at about the state of Washington eastward across the Rockies following roughly the Canadian boundary and leaving continental North America a very narrow area in the vicinity of the New England states. This approximates the Van Cleeve storm track map. It showed clearly that there was a greater demand for adjustment of the human body to cyclonic storms at Duluth than at San Francisco. Meteorologically speaking, each three or four days a meteorological episode crosses in this cyclonic area, traveling eastward across the United States. Meteorological episode is a term applied to a cold area traveling eastward with a warm area between it and the cold area still further to the east.

He showed clearly by curves, based on the deaths occurring in representative states, and in the United States as a whole, that this death rate varies directly with the barometric pressure adjustment required. This is particularly noticeable at the time of the equinox in the spring and the fall. The barometric variation is the greatest at these periods and the death rate correspondingly great. The birth rate is inversely to the death rate at these periods. He traced the variation of disease phenomenon in certain patients afflicted with paralysis, pneumonia, diabetes, and insanity, showing a definite relationship between barometric pressure and the well-being and body chemistry of these patients.

New A.C. Assn. Meets

CHICAGO—The third meeting of the Chicago Air Conditioning Association since the new group was organized was held March 26 and about forty representatives of companies interested in the industry were present. A report of the board of directors was given concerning its recent meetings called to aid in coordinating various groups.

A general discussion on the industry's problems followed the business meeting. Those who participated were: P. R. West, president, Ventilating and Air Conditioning Association; R. D. Wolff, consulting engineer and past president of the ASHVE Chicago Chapter; C. O. Frary, Bryant Heater Co.; W. J. O'Neil, Iron Fireman Mfg. Co.; R. B. Hayward, contractor; C. D. Andel, Andel & Co.; J. H. Manny, Robinson Furnace Co., and M. K. Arenburg, Ilg Electric Ventilating Co.



Speakers at the contractors' meeting of the "Radiator Heat with Air Conditioning" campaign held in Chicago, March 24, at the Sherman Hotel. Left to right: Louis T. Braun, executive secretary, Chicago Master Steamfitters' Assn.; C. G. Newton, vice-president, Stevens-Root Co., chairman, Chicago Committee, National Bureau of Heating & Air Conditioning; W. J. Spillane, J. B. Clow & Co., chairman, Wholesalers' and Manufacturers' Division of the campaign; Kenneth MacKenzie, manager, Chicago Branch, American Radiator Co., committee member, and J. L. DeNeille, president, Heating, Piping and Air Conditioning Contractors National Association.

Foil Insulation Described

TORONTO, ONT.—Fifty members of the Ontario Chapter, ASHVE, attended the February meeting held in the clubrooms. Features of the affair were two interesting addresses, one by D. N. McIntosh, engineer, Brantford Roofing Co., Brantford, who described the use of aluminum foil for insulation purposes. Allyne Rooke, engineer, Page-Hersey Tubes Ltd., Welland, outlined in an interesting manner the manufacture of steel pipe and tubing.

N. Y. Heating Contractors Hold Second Annual Dinner Dance; Weare New Executive Secretary

NEW YORK—The Association of Heating Contractors of Greater New York, Inc., held its second annual dance and entertainment at the French Casino, March 3. About 500 attended. The dance was in charge of an entertainment committee consisting of B. Spiegel, E. Gruen, M. M. Weiss, J. C. Weare, and S. Dickstein.

J. Miller

P. Wolf

J. C. Weare

F. W. Hanburger

The association was formed early in 1933 and has as its aims the promotion of the welfare of persons in the heating industry; elevation of the general standards of conduct of business, and the fostering and maintenance of good fellowship and friendship among business men in the heating trade.

The association has grown to such an extent that the executive work in connection with the management of the association's affairs became heavy enough so that it was recently decided to engage the services of an executive secretary, and John C. Weare was selected for this position. Mr. Weare has been closely identified with the real estate industry for a number of years, and during the war was a commissioned officer in the U. S. Navy. Headquarters of the association are at 1 East 125 St., New York.

Prof. F. W. Hanburger, City College of New York, and the only honorary member of the association, is chairman of the membership committee. According to Professor Hanburger, any legitimate heating contractor transacting business in Greater New York is eligible for membership in the association.

The association will hold its open meeting April 28 at the Hotel Capitol, 8 Ave. and 51 St. Admission will be by invitation only. One of the purposes of this meeting will be to allow manufacturers and jobbers, especially those who have supported the association, and the members of the association, to discuss business. This meeting is thus distinguished from purely social meetings where conversations regarding business are more or less barred.

Officers of the association are as follows: J. Miller, Universal Heating Co., president; P. Wolf, City Contracting Co., vice-president; J. C. Weare, secretary; J. Kessler, Jacler Heating Co., treasurer, and M. M. Weiss, financial secretary.

COMING EVENTS

APRIL 6-11, 1936. *Southern Industrial Show, Textile Hall, Greenville, S. Car.*

APRIL 14-18, 1936. *Twelfth National Oil Burner Show, Convention Hall, Detroit, Mich.*

APRIL 20-24, 1936. *Midwest Power Engineering Conference (Palmer House) and Engineering and Power Exposition (International Amphitheatre), Chicago, Ill.*

APRIL 21-23, 1936. *Conference on Solid Fuels and Domestic Stokers, University of Wisconsin, Department of Mechanical Engineering, Madison, Wis. Communicate with Prof. L. A. Wilson, Mechanical Engineering Building of the University.*

MAY 12-16, 1936. *First Wisconsin Air Conditioning and Heating Exposition, Milwaukee Auditorium, Milwaukee, Wis.*

MAY 14-16, 1936. *Spring Meeting of Refrigerating Machinery Association, Hot Springs, Va.*

MAY 16, 1936. *Annual Meeting of Air Conditioning Manufacturers' Association, Hot Springs, Va.*

MAY 25-27, 1936. *Forty-Seventh Annual Convention of Heating, Piping and Air Conditioning Contractors National Association, Bellevue-Stratford Hotel, Philadelphia, Pa.*

JUNE 9-11, 1936. *Short Course in Coal Utilization, Department of Mining and Metallurgical Engineering, University of Illinois, Urbana, Ill.*

JUNE 16-19, 1936. *National District Heating Association Convention and Exhibit, Pantlind Hotel, Grand Rapids, Mich.*

JUNE 17-18, 1936. *National Warm Air Heating and Air Conditioning Association Convention, Deshler-Wallick Hotel, Columbus, Ohio.*

JUNE 17-20, 1936. *Semi-Annual Meeting of the American Society of Mechanical Engineers, Hotel Adolphus, Dallas, Tex.*

Arco Holds Sales Conventions

NEW YORK—A nationwide campaign on air conditioning was opened last month by American Radiator Company to establish the radiator heating contractor as the source of conditioning equipment.

Two conventions for all sales representatives, the first that the company has held in four years, marked the opening of the campaign which Arthur R. Herske, vice-president and general manager of sales, declared would bring seven advertisements from the press, to be distributed "every time the clock ticks every second of the day and night."

Held at the New Yorker Hotel in New York, March 6-7, and at the Stevens Hotel in Chicago, March 13-14, the conventions, which lasted for two days each, served as intensive sales training courses to prepare the company's salesmen in aiding the radiator heating contractor to present "the truth in air conditioning to the American public."

The company will definitely adopt a policy of presenting the radiator heating contractor as the source not of equipment for either heating or air conditioning but as the source of integral conditioning systems, the purpose of which is complete and absolute control of indoor comfort.

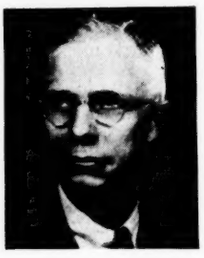
Contractors to Attend Sales Clinic at Convention

NEW YORK—The 47th annual convention of the Heating, Piping and Air Conditioning Contractors National Association will be held at the Bellevue-Stratford, Philadelphia, May 25-27. At the meeting a report will be presented on the Chicago campaign for the promotion of air conditioning with radiator heat.

A sales clinic will be held Monday May 25, when the sales problems of the industry will be discussed from the standpoints of the manufacturer, wholesaler, and contractor.

New Heating Department

PITTSBURGH—A heating department has been organized by the Chandler-Boyd Supply Co., to be under the direction of Eugene Stitt, well known in this territory.



Eugene Stitt

The department will be special stocking distributor for C. A. Dunham Co. equipment, and will also maintain a complete line of heating equipment.

Airtemp New York Sales Corporation, New York distributor for Chrysler Motors air conditioning products, has appointed **B. S. Williams** vice-president and general manager. Mr. Williams was formerly with York Ice Machinery Corporation.

B. S. Williams

Carrier Engineering Corp., Newark, N. J., has consolidated at its home office personnel responsible for engineering, installation, and service on contract air conditioning work for the eastern sales territory. The new set-up centers in Newark the engineering and construction service formerly performed by district offices in New York and Philadelphia.

W. M. Chace Company, Detroit, has completed its third plant expansion in ten years. While it was necessary to

Recently enlarged Chace plant.

double the size of the plant in 1934, due to the rapid increase in the use of Chace thermostatic bimetal in temperature control devices, last year it again became necessary to begin a large plant expansion. The recently completed addition increases floor space 50%. The additional quarters have been equipped with new manufacturing equipment.

Delco-Frigidaire Corp., Dayton, has appointed **Elliott-Lewis Co.,** as Philadelphia dealer for its line of air conditioning equipment. In taking over the franchise, Elliott-Lewis has added a staff of air conditioning engineers to its staff of heating engineers, headed by **J. E. Rowland,** general manager.

General Refractories Co., Philadelphia, has appointed **Braman, Dow & Co.,** Boston, as dealer agents in that area.

Holcomb & Hoke Mfg. Co., Indianapolis, Ind., has announced the addition of the following distributors to its FireTender Stoker Division: **Brown Supply Co.,** St. Louis, Mo., and **General Distributing Corp.,** Saginaw, Mich.

New dealers announced by the company are: **Clean Heat Co.,** Columbus, Ohio: **Howard Feaser,** Niles, Mich.; **C. B. Wilt,** Salem, Ind.; **Broadway Lumber Co.,** Greenfield, Ind.; **Frank Lane,** French Lick, Ind.; **Otis Burton,** Orleans, Ind.; **A. D. Burton,** Bedford, Ind.; **Lasiter Bros.,** Greenwood, Ind.; **J. F. Giddings,** North Vernon, Ind., and **The Fixit Shop,** Kalamazoo, Mich.

Independent Air Filter Co., Inc., 1 North La Salle St., Chicago, has opened a New York office at 55 West 42 St., New York. **Harry Gitterman** is the representative in charge.

Iron Fireman Manufacturing Co., Portland, Ore., announces completion by May 1 of an expansion program which will increase by 50% the capacity of its Cleveland factory. The company is erecting two large additions. The first is a steel frame building 172 × 211 ft. long.

Kelvinator Corp., Detroit, has added 26 new distributors to handle its line of air conditioning equipment. A few of these follow: William F. Batterham Co., B. F. Gilbert Co. Inc., and Sam S. Glauber, Inc., New York; Wittenmeier Machinery Co., Chicago; Sigwalt Lumber Co., Des Plaines, Ill.; Collin & Spalding, Elgin, Ill.; Waukegan Refrigeration Co., Waukegan, Ill.; John A. Portner, Wheaton, Ill.; Stucky Brothers Inc., Fort Wayne, Ind.; Lambert-Grisham Hardware Co., Henderson, and Owensboro, Ky.

The Mercoid Corp., Chicago, announces that **Lewis B. Reed** has been elected to the office of president of the corporation, succeeding the late

Lewis B. Reed

L. H. Van Ness. Mr. Reed has been associated with the company since its beginning, but was not actively engaged in its management until December, 1930, when he became vice-president.

Minneapolis-Honeywell Regulator Co., Minneapolis, Minn., plans by advertising to create greater interest on the part of consumers, large space operators, and industry in the subject of automatic heating and air conditioning, according to George B. Benton, advertising manager. Starting in January with page and two column copy in five consumer magazines, this campaign stresses the general theme "Be Modern — Install Automatic Heating and Air Conditioning."

This campaign will be followed by one using the general theme—"Air Conditioning is Knocking at Your Door." This will stress the fact that air conditioning will pay big dividends in health and comfort to the home-owner and reward liberally the hotel, apartment, or theater owner in increased patronage and satisfied customers. Industry is also reminded that year 'round air conditioning improves plant processing and employe efficiency. This campaign will continue through March, April, June, and July.

A further effort in selected territories selling the public on the urgency of "Installing Automatic Heat Now," will be launched in August, to stimulate fall business and increased oil burner, gas burner, and stoker sales. There will also be a magazine campaign urging the public again to install air conditioning with emphasis on winter humidification.

In connection with the campaign, the company has published an informative booklet entitled, "This Thing Called Automatic Heating and Air Conditioning," designed to clear up current misunderstandings and misconceptions, especially about air conditioning.

National Radiator Corp., Johnstown, Pa., has opened a warehouse in Newark, N. J., to supplement the facilities afforded by its warehouse at Long Island City.

Penn Electric Switch Co., Des Moines, Iowa, has moved its New York office from 15 Laight St. to **101 Park Ave.**

Penn Electric Switch Co., Des Moines, Iowa, has named **R. D. Marshall** as midwestern representative, in charge of sales in the area served by the Des Moines office.

This company has also named **Melchior, Armstrong, Dessau Co.,** 300 Fourth Ave., New York, as distributor of its controls for oil, gas, and stoker heating. They will continue to handle Penn refrigeration and air conditioning controls. **Preferred Utilities Manufacturing Co.,** 33 West 60 St., New York, continues as distributor of Penn heating controls.

Youngstown Sheet & Tube Co., Youngstown, Ohio, announces that **Harold R. Goodwin,** widely known in the Detroit territory where he has been engaged in the sale of plumbing materials for 20 years, has joined the company's sales organization at Detroit. Mr. Goodwin was associated with the Murray W. Sales Co., Detroit plumbing jobber, from 1917 to 1924, and with the Kimball-Eisenberg Co., engaged in the same field, from 1924 until recently.

NEW EQUIPMENT

Pretty Neat Drawing Board

NAME—"Pretty Neat" drawing board.

PURPOSE—A light weight and attractive appearing drawing board for use at the desk or elsewhere, but especially applicable for carrying in a brief case.

FEATURES—Made of pressed wood. Paper is held by strong clamps in the corners and can be quickly inserted or removed and needs no thumb tacks. Clamps are lifted by pressure of finger on small buttons on back of board. Edges are all square so that triangle can be used along any edge, thus eliminating

necessity for T-square. Made in several sizes and weighs less than one pound. Designed for sketches and charts and other drawings, and is especially useful for sales engineers and field engineers where a larger board is too cumbersome to carry around.

SIZES AND CAPACITIES—Available in sizes to take the following size paper: 8½ x 11, 9 x 12, and 10 x 15 in. Other sizes made to order. Price $1.50 f.o.b. Riverside, Calif., for 8½ x 11 in. size. Prices for the larger sized boards are $1.75 and $2.25.

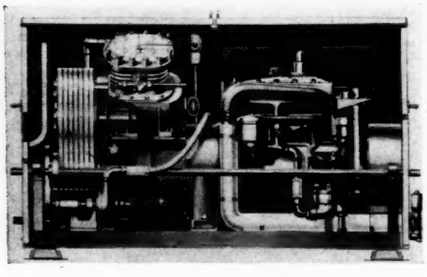
MADE BY—*H. E. Twomley, 7154 Magnolia Ave., Riverside, Calif.*

Waukesha Gas Engine Compressor

NAME—Waukesha gas operated "ice" engine.

PURPOSE—For summer air conditioning.

FEATURES—A refrigeration machine for heavy duty air conditioning which is driven by a gas engine, using manufactured or natural gas. The compressor and engine are built as a single unit. Both are four cylinder, the



latter being of the V type. Operation is automatic. Gas unit can be operated in connection with a tap water supply for condensing on the low-pressure side of the air conditioning system which can be connected to an evaporative type of condenser.

SIZES AND CAPACITIES—Equivalent to 7 tons of refrigeration effect. Engine has 20 available h.p. Compressor runs at 600 r.p.m., uses Freon or methyl chloride refrigerant. Unit enclosed in sound insulated cabinet measuring 56 x 27 x 35 in. Weight, 1,000 lb.

MADE BY—*Waukesha Motor Co., Waukesha, Wis.*

L-R Flexible Coupling

NAME AND MODEL NUMBER — L-R Type W flexible coupling.

FEATURES — Individual load cushions are free floating between the metal jaws and rest upon the central hub, and are held in place by a spiral steel spring. Cushion available in three types of materials, one being a brake lining material, a second of belting leather, and a third of rubber duck fabric.

SIZES AND CAPACITIES—In standard size with bores from 3 to 14 in.

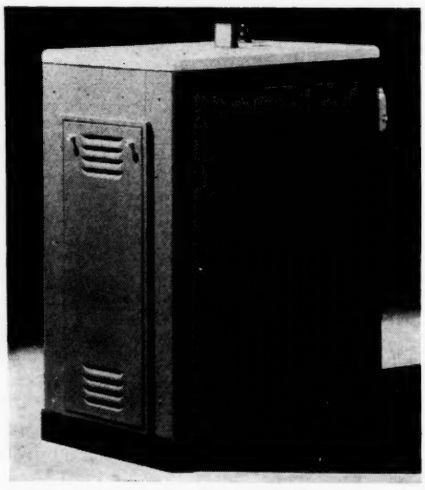
MADE BY—*Lovejoy Tool Works, Chicago.*

Pierce Oil Boiler

NAME—Pierce oil boiler.

PURPOSE—Designed for use with, but does not include, oil burner.

FEATURES—Has water leg construction so that combustion chamber is completely surrounded by water. Products of combustion make two passes before being drawn into the three-way indirect heating surface passages. Finished in two-toned gray enamel jacket backed by asbestos and sound-proofing material. Has refractory-lined Pyrex glass inspection door.



SIZES AND CAPACITIES—Available in six sizes with steam radiator loads ranging from 570 to 1520 sq. ft., and for minimum burner capacities of 1.22 to 3.23 gal. per hr.

MADE BY—*Pierce, Butler Radiator Corp., Syracuse, N. Y.*

Robeson Air Conditioner

NAME—Robeson warm air type air conditioner.

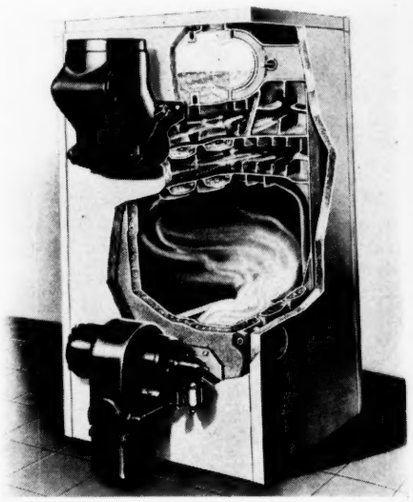
FEATURES—The manufacturer claims the unit is smallest in size for any manufactured in equal capacity. A feature of the unit is the mounting of the blower in the space above the combustion chamber. The unit includes filters, blower, humidifier, and economizer. Designed for oil burning, it is available in four models with capacities ranging from 110,000 to 300,000 B.t.u. per hr. at the register.

MADE BY—*Robeson Engineering Co., Inc., East Orange, N. J.*

Timken Boiler Burner

NAME AND MODEL NUMBER—GC Boiler-burner unit.

FEATURES—Moving parts said to have been



reduced to a minimum for pressure type burners, has form control device for controlling of air supply.

SIZES AND CAPACITIES—Five sizes ranging from 510 to 1,090 sq. ft. of steam radiator.

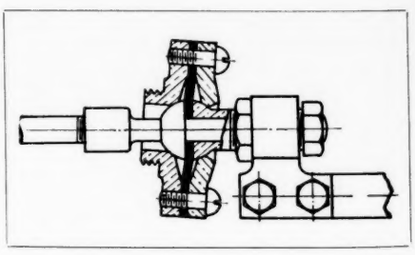
MADE BY—*Timken Silent Automatic Co., Detroit, Mich.*

Davis Float Box Construction

NAME—Packless float box construction.

PURPOSE—Available on all of the company's float boxes for valve, switch, signal, or indicating purposes.

FEATURES—Designed primarily for oil and chemical industries where highly volatile fluids are handled. It also offers protection against leakage and maintenance nuisance wherever water or other liquids are stored under pressure. Elimination of conventional packing is accomplished by a flexible diaphragm, a ball joint construction which forms



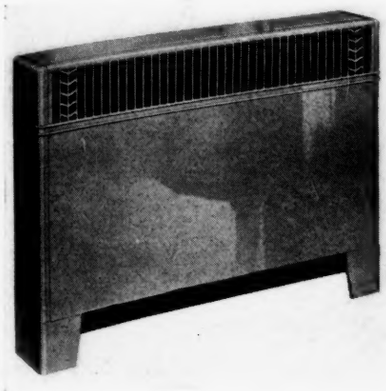
a fixed seal. Pressure thrust is carried by ball joint, the outboard side of which is lubricated with long-life graphite grease.

MADE BY—*Davis Regulator Co., Chicago.*

Modine Copper Convector

NAME—Modine copper convector.

FEATURES—Available in a new line designed for improved appearance by a furniture stylist. A wide selection of grille design is available to architects who wish to make up their own patterns to harmonize with the building interior. Various shades and tones are available in the enclosures. Rounded corners are used. To protect corners of the heating element, cornice strips are metallically bonded to the fins on the top, bottom, and on both sides. Unit is said to provide greater heating capacity than the old model, size for size. Company announces a new catalog on



this line now available in which considerable space is devoted to the use of this product with air conditioning.

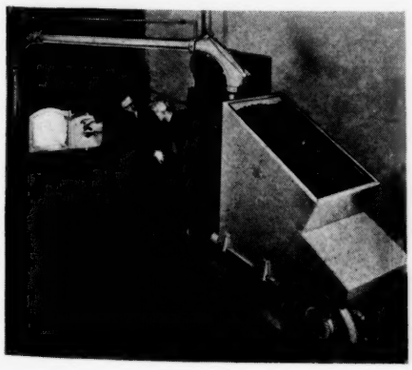
MADE BY—*Modine Manufacturing Co., Racine, Wis.*

Iron Fireman Pneumatic Spreader

NAME—Pneumatic spreader.

PURPOSE — For spreading coal over the grates of a boiler; said to combine best features of overfeed stoker and powdered coal burner.

FEATURES — Fines and lump coal up to 1½ in. are carried into furnace on the stream of air. Finer particles burn in suspension, with flame similar to that produced by a powdered coal burner. Larger pieces burn in a shallow layer on the grates. Is not necessary to rebuild boiler front; the cutting of a hole in the boiler front large enough to admit the spreader nozzle is sufficient. Particularly



adapted to firing of power and process boilers for industrial uses.

SIZES AND CAPACITIES—In five sizes, with capacities ranging from 75 to 500 boiler hp., and in either hopper or bunker models.

MADE BY — *Iron Fireman Manufacturing Co., Portland, Ore.*

Synchron Control

NAME—Synchron control unit.

PURPOSE—Time control unit for oil or gas burners for reducing length of off periods.

FEATURES—Device is set to feed fuel at the rate of 30 sec. to 5 min. out of every quarter-hour, depending on the size of the heating plant. Manufacturer claims that use of this control increases baseboard temperature 4° without increasing the 70° temperature at the breathing line. Operates independent of room thermostat and can be cut in during cold weather and cut out in milder weather. It can be installed with any make temperature control system, two or three wire, line voltage, or low voltage. Cams operate make and break system. Contact point made of tungsten.

SIZES AND CAPACITIES — Furnished in heavy steel box, 6½ x 4½ x 3½ in.

MADE BY—*Industrial Engineering Corp., Evansville, Ind.*

General Controls Magnetic Valve

NAME AND MODEL NUMBER—Type K-10 magnetic valve.

PURPOSE — Control of fluids and gases at high pressures.

FEATURES—Solenoid type with a lever action to exert high seating pressure. Valve closes with flow. Can be used for viscous fluids as well as water, gas, air, refrigerants, steam, and heavy oils. Operation is satisfactory on pressures up to 300 lb. on small port open-

ings. Consumes 14 watts. Is of packless design with internal parts of stainless steel and brass. Waterproof coils are removable and may be rotated in any direction for ease of wiring.

MADE BY — *General Controls Co., Los Angeles, Calif.*

Surface and Needle Pyrometers

NAME—Combination surface and needle pyrometers.

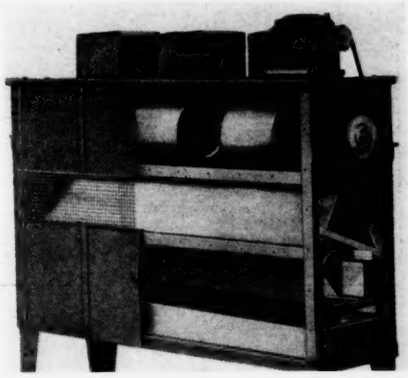
FEATURES—By interchanging various types of thermocouples, said to be a matter of a few seconds, the instruments can be made into four different types of surface and needle pyrometers. Indicator is equipped with clamping device whereby contact and reading of actual temperatures do not have to be made simultaneously as the indicating needle will stay fixed at the correct temperature indication.

MADE BY—*The Pyrometer Instrument Co., New York, N. Y.*

Young Unit Heaters, Model FH

NAME AND MODEL NUMBER—Model FH unit heaters.

FEATURES—Made for floor, vertical wall, or horizontal ceiling mountings. Direction of air flow from each nozzle can be governed by changing the position of nozzle on the top panel of the unit heater so that the air can be circulated in three or four directions from the same heater if desired. Casing is of heavy gage steel panels, thoroughly braced and ribbed. Copper extended surface heating element. Tubes are rolled into one-piece, nickel alloy, cast iron header tanks. Heating element is designed for working pressures up to 200 lb. per sq. in.



Blowers are mounted on heavy shaft and driven by ball bearing motor through V-belt drive. Motor mounted on adjustable bracket to regulate belt tension. Can be equipped with Thermadjust dampers, which has the advantage of being able to bypass around the heating element all or any of air being handled. Bypass dampers can be manually controlled or can be furnished to operate with a damper motor thermostatically controlled.

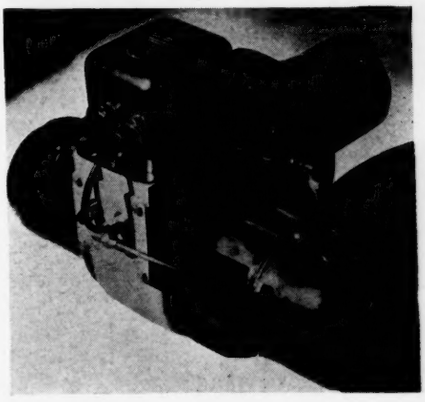
SIZES AND CAPACITIES—Available in 10 models with four sizes of motors for each, giving a range of capacity from 118,700 to 802,000 B.t.u. per hr. when using 2-lb. steam.

MADE BY—*Young Radiator Co., Racine, Wis.*

Bethlehem-Doe Flange Type Oil Burner

NAME AND MODEL NUMBER—Model FE flange type oil burner for enclosed oil burning boilers.

FEATURES—Has monitor type motor-fan-pump unit which can be removed bodily for



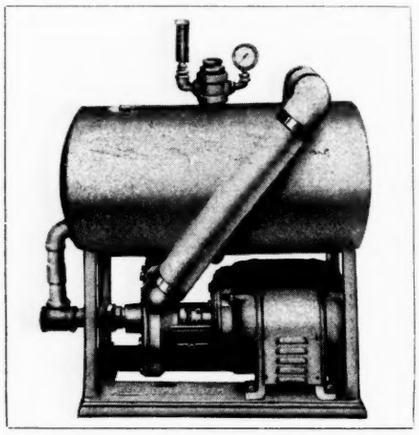
servicing. Also has live rubber cradle mounting for support of pump unit. Flame diffuser is of the company's Rotaire diffuser type.

MADE BY—*The Bethlehem Foundry & Machine Co., Bethlehem, Pa.*

Adsco-Becker Condensate Return Unit

NAME—Adsco-Becker condensate and steam return system.

PURPOSE—Returning steam and condensate to the boiler at high temperatures in applications in industrial plants and buildings where process work is carried on.



FEATURES—The device is one piece of equipment, motor driven with one moving part.

MADE BY—*American District Steam Co., North Tonawanda, N. Y., east of the Mississippi and by the Becker Pump Co., Oakland, Calif., in the West.*

Bethlehem-Doe Conversion Oil Burner

NAME AND MODEL NUMBER—Model CE conversion oil burner.

FEATURES—Has monitor type motor-fan-pump unit which can be removed bodily for servicing. Also has live rubber cradle mounting for support of pump unit. Flame diffuser is of the company's Rotaire diffuser type.

MADE BY—*The Bethlehem Foundry & Machine Co., Bethlehem, Pa.*

Ortho-Clime Air Conditioners

NAME—Additions to the Ortho-Clime air conditioning units.

FEATURES—Model 612 is designed for both cooling and heating. Has a 1-ton cooling capacity. Two ceiling suspended units, models 715 and 736, are available in 1½-ton and 4-ton cooling capacities. A fourth one is a self-contained room cooler of about ¾-ton cooling capacity.

MADE BY—*Fairbanks, Morse & Co., Chicago.*

Prest-O-Lite Halide Leak Detector

NAME—Prest-O-Lite Halide leak detector.

PURPOSE—For detecting refrigerant leaks, especially where odorless, tasteless, and colorless refrigerants are used.

FEATURES—Insurance companies prohibit testing for leaks of a combustible gas with any flame. The Halide leak detector consists of a needle valve torch handle assembly, a burner which includes a suction nipple for attaching a rubber hose, and a chimney with a copper reaction plate. The rapid flow of acetylene through the burner causes a suction which draws in any refrigerant gas near the open end of the suction tube.

DETAILS OF OPERATION—To operate the detector, the flame is first adjusted so that the top of the outside cone is level with or slightly above the chimney of the detector. The open end of the suction tube is used to explore around those places where a leak might occur. Any halide refrigerant gas drawn into the burner is decomposed with the formation of free acids. These acids, coming into contact with the hot copper reaction plate, cause instant color response in the flame. Visible color indication of the smallest concentration of refrigerant gas by a green-tinted flame is given by the detector. When a large amount is present, the flame assumes an intense violet color. After the source of the leak has been passed, the flame clears almost instantly. The two-color variation of flame gives visible indication of the amount of refrigerant gas present.

MADE BY—*The Linde Air Products Co., New York.*

Foxboro Remote Control

NAME—Remote hand control.

PURPOSE—To operate manually from a central location a valve as far as 700 ft. away.

FEATURES—Control is mounted on a centrally located panel, and from the panel the operator can reset a valve to control temperature, pressure, flow, or liquid level using readings from his recording instruments to guide him in making the correct adjustments. Valve opening can be altered as little as two-thousandths of an inch, and when used in conjunction with a Stabilflo valve the controlled flow is said to be changed by less than 1%. Finished in lacquered cast bronze trimmed with nickel.

MADE BY—*The Foxboro Co., Foxboro, Mass.*

Balsam-Wool Duct Insulation

NAME—Balsam-wool duct insulation.

PURPOSE—For sound and heat insulation of ducts. Consists of a wood fiber mat, chemically treated to be fire, moisture, vermin, and rot resistant. Mat is covered on each surface with a Kraft liner. One long edge is sealed, the other is sealed and flanged. It is applied to the exterior of the duct and fastened by an adhesive. Advantages claimed by the manufacturer include the following: Duct is not punctured by screws or other fastening devices; wide strips of insulation eliminate many joints; all joints can be sealed; application cost said to be low as result of wide strips and because the material can be wrapped around the duct.

SIZES AND CAPACITIES—Available in either ½ or 1 in. thicknesses, and in widths of 21 and 40 in. Special widths up to 96 in. obtainable. Properties of the ½ and 1 in. insulation, respectively, are: weight in pound per square foot, 0.24 and 0.37; conductance in B.t.u. per square foot per hour per degree temperature difference, 0.394 and 0.218.

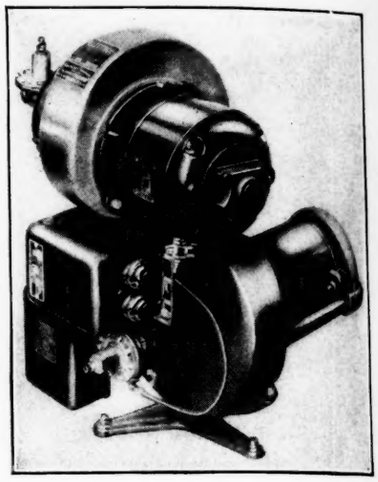
MADE BY—*Wood Conversion Co., St. Paul, Minn.*

Electrol TCV Oil Burner

NAME AND MODEL NUMBER—TCV oil burner.

PURPOSE—For jacketed boilers.

FEATURES—Embodies all Electrol characteristics such as curved air chamber which gives the air for combustion a rotating motion, creating a ball flame; master control; cut-off



valve at nozzle and electric flame ignition. Vertically mounted.

MADE BY—*Electrol Incorporated, Clifton, N. J.*

Hancock Globe Valves

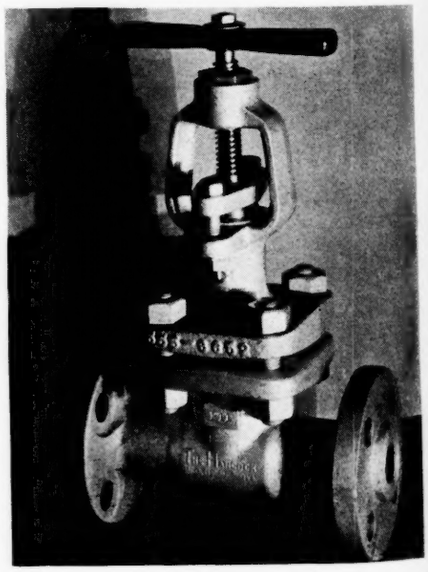
NAME—Hancock forged steel globe valves.

PURPOSE—For pressures up to 1,500 lb.

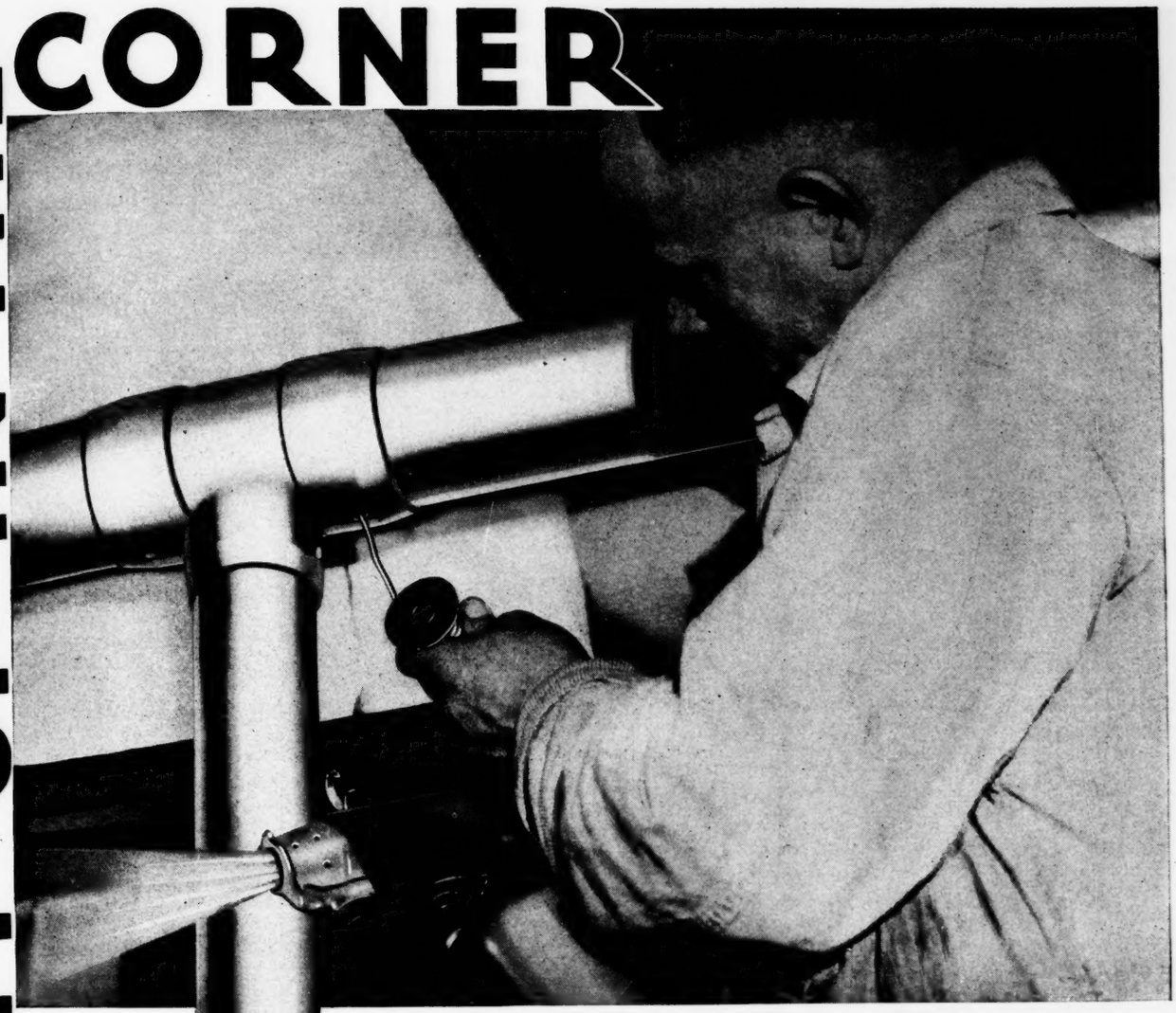
FEATURES—Designed with tongue and groove bolted bonnet for high temperatures. Valve seat ring and disc made of heat treated stainless steel. Long condensation chamber protects and increases life of packing. Wearing parts are renewable. Valves said to be easy to repack in service because of stainless steel back seat and because gland bolts come completely out allowing ample finger room and because of a relief plug for bleeding condensation chamber while repacking.

SIZES AND CAPACITIES—A range of sizes in either cone or plug type.

MADE BY—*The Hancock Valve Co., Bridgeport, Conn.*



RIGHT UP IN THE CORNER



Asbestos Sheet Used Back of Job Prevents Scorching

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One of the many advantages of STREAMLINE products is the fact that they require a very minimum of space. STREAMLINE Fittings, although producing enormously strong joints, are very little larger than the pipe lines which they connect. They do not protrude like screw type fittings. They can be readily installed close up to walls, ceilings, or partitions and in places otherwise inaccessible, because no room is required for wrench grip or wrench swing. The fitting is located exactly where required and soldered.

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Absence of clogging rust; freedom from the harmful effects of vibration; exceptionally low heat loss by radiation and, above all, economy of operating costs and quick transference of the heating element to the points of radiation are but a few of the many reasons why an installation of STREAMLINE Copper Pipe and Fittings is ideal for a trouble-free, efficient heating system.

Send for our Catalog F and Solder Instruction Card

STREAMLINE PIPE AND FITTINGS

PORT HURON, CO. MICHIGAN

DIVISION OF MUELLER BRASS CO.

Degree-Days for February, 1936

Continuing HEATING & VENTILATING's eighth year of publishing degree-day data for various large cities in the United States. This month, seven new cities—Albany, Cheyenne, Erie, Lincoln, Little Rock, Madison, and Utica—have been added, bringing the total number of cities reported up to 63.

	Albany, N. Y.	Atlanta, Ga.	Baltimore, Md.	Birmingham, Ala.	Boston, Mass.	Buffalo, N. Y.	Burlington, Vt.
Degree-days for February, 1936.....	1352	678	1022	622	1224	1348	1495
Degree-days, Sept. 1, '35 to Feb. 29, '36	5058	2789	3780	2541	4602	5214	6046
Degree-days, Sept. 1, '34 to Feb. 29, '35	4915	2340	3355	1994	4582	4781	5827
Degree-days, Sept. 1 to Feb. 29, Normal	4867	2370	3462	2079	4322	4767	5601

	Chevenne, Wyo.	Chicago, Ill.	Cincinnati, Ohio	Cleveland, Ohio	Columbus, Ohio	Denver, Colo.	Des Moines, Iowa
Degree-days for February, 1936.....	1301	1444	1108	1275	1179	1104	1652
Degree-days, Sept. 1, '35 to Feb. 29, '36	5441	5309	4470	4813	4678	4387	5965
Degree-days, Sept. 1, '34 to Feb. 29, '35	4473	4362	3561	4136	3862	3478	4626
Degree-days, Sept. 1 to Feb. 29, Normal	5178	4582	3666	4440	4113	4273	4935

	Detroit, Mich.	Dodge City, Kan.	Duluth, Minn.	El Paso, Tex.	Erie, Pa.	Evansville, Ind.	Fort Wayne, Ind.
Degree-days for February, 1936.....	1420	1198	1943	397	1323	1053	1387
Degree-days, Sept. 1, '35 to Feb. 29, '36	5320	4337	7706	2026	4945	4045	5308
Degree-days, Sept. 1, '34 to Feb. 29, '35	4233	3273	6644	1782	4366	3077	4415
Degree-days, Sept. 1 to Feb. 29, Normal	4719	3958	6634	2101	4426	3348	4512

	Grand Rapids, Mich.	Harrisburg, Pa.	Hartford, Conn.	Indianapolis, Ind.	Kansas City, Mo.	La Crosse, Wis.	Lincoln, Neb.
Degree-days for February, 1936.....	1447	1186	1253	1259	1313	1757	1612
Degree-days, Sept. 1, '35 to Feb. 29, '36	5333	4442	4618	4854	4730	6439	5629
Degree-days, Sept. 1, '34 to Feb. 29, '35	4658	3999	4500	3850	3591	5446	4182
Degree-days, Sept. 1 to Feb. 29, Normal	4863	4077	4416	4099	4216	5579	4696

	Little Rock, Ark.	Los Angeles, Calif.	Louisville, Ky.	Madison, Wis.	Memphis, Tenn.	Milwaukee, Wis.	Minneapolis, Minn.
Degree-days for February, 1936.....	822	251	1027	1684	825	1573	1886
Degree-days, Sept. 1, '35 to Feb. 29, '36	3060	769	4074	6253	3151	5680	7010
Degree-days, Sept. 1, '34 to Feb. 29, '35	2267	723	3168	5246	2271	4859	5718
Degree-days, Sept. 1 to Feb. 29, Normal	2380	1016	3327	5556	2470	5185	5950

	Nashville, Tenn.	New Haven, Conn.	New Orleans, La.	New York, N. Y.	Norfolk, Va.	Oklahoma City, Okla.	Omaha, Neb.
Degree-days for February, 1936.....	851	1194	332	1110	811	916	1702
Degree-days, Sept. 1, '35 to Feb. 29, '36	3440	4443	1297	4080	2971	3433	5976
Degree-days, Sept. 1, '34 to Feb. 29, '35	2617	4263	876	3893	2548	2488	4293
Degree-days, Sept. 1 to Feb. 29, Normal	2922	4230	965	3870	2583	2986	4765

	Peoria, Ill.	Philadelphia, Pa.	Pittsburgh, Pa.	Portland, Me.	Portland, Ore.	Providence, R. I.	Reading, Pa.
Degree-days for February, 1936.....	1455	1082	1142	1305	904	1206	1157
Degree-days, Sept. 1, '35 to Feb. 29, '36	5402	3940	4431†	5139	3334	4535	4309
Degree-days, Sept. 1, '34 to Feb. 29, '35	4279	3568	3816	5212	2734	4437	3934
Degree-days, Sept. 1 to Feb. 29, Normal	4673	3629	3947	4985	3174	4315	4082

	Richmond, Va.	Rochester, N. Y.	St. Louis, Mo.	Salt Lake City, Utah	San Francisco, Calif.	Scranton, Pa.	Seattle, Wash.
Degree-days for February, 1936.....	884	1341	1181	902	337	1260	834
Degree-days, Sept. 1, '35 to Feb. 29, '36	3464	5147‡	4410	4239	1653	4932	3254
Degree-days, Sept. 1, '34 to Feb. 29, '35	3060	4765	3294	3773	1522	4520	2875
Degree-days, Sept. 1 to Feb. 29, Normal	2933	4825	3652	4144	1770	4552	3309

	Spokane, Wash.	Syracuse, N. Y.	Toledo, Ohio	Trenton, N. J.	Utica, N. Y.	Washington, D. C.	Wichita, Kan.
Degree-days for February, 1936.....	1384	1327	1372	1162	1425	1027	1162
Degree-days, Sept. 1, '35 to Feb. 29, '36	5072	5168	5217	4304*	5499	3825	4263
Degree-days, Sept. 1, '34 to Feb. 29, '35	4190	4827	4449	3960	5057	3411	3211
Degree-days, Sept. 1 to Feb. 29, Normal	4707	4980	4465	3715	4966	3559	3773

To obtain unit fuel consumption figures for any city shown above, multiply the number of degree-days by the following factors: for coal, 0.008; for oil, 0.00068; for gas, 0.096. Figures obtained will show coal consumption in pounds per square foot of radiator surface; oil consumption in gallons per square foot; gas consumption in cubic feet per square foot—all for the period covered by the number of degree-days. The figures assume the use of steam radiators emitting 240 B.t.u. per sq. ft. per hr., a system operation at 100% efficiency, and radiators calculated for maintaining 70° in zero weather. Heating values assumed for the fuels are 12,000 B.t.u. per lb. for coal; 140,000 B.t.u. per gal. for oil, and 1,000 B.t.u. per cu. ft. for gas. To correct for other heating values, efficiency and design conditions, follow the method explained in the "HEATING & VENTILATING Degree-day Handbook" or on H. & V.'s Reference Data Sheet Nos. 67 and 68. Degree-days as given above for "a normal" month or season are based on averages for a long period of years, ending about 1922. Averages covering different periods will disagree with the above figures slightly.

Revisions: †Pittsburgh degree-days for January should have been 1206 instead of 1315. ‡Rochester degree-days for January should have been 1289 instead of 1290. *Trenton degree-days for January should have been 1154 instead of 1155.

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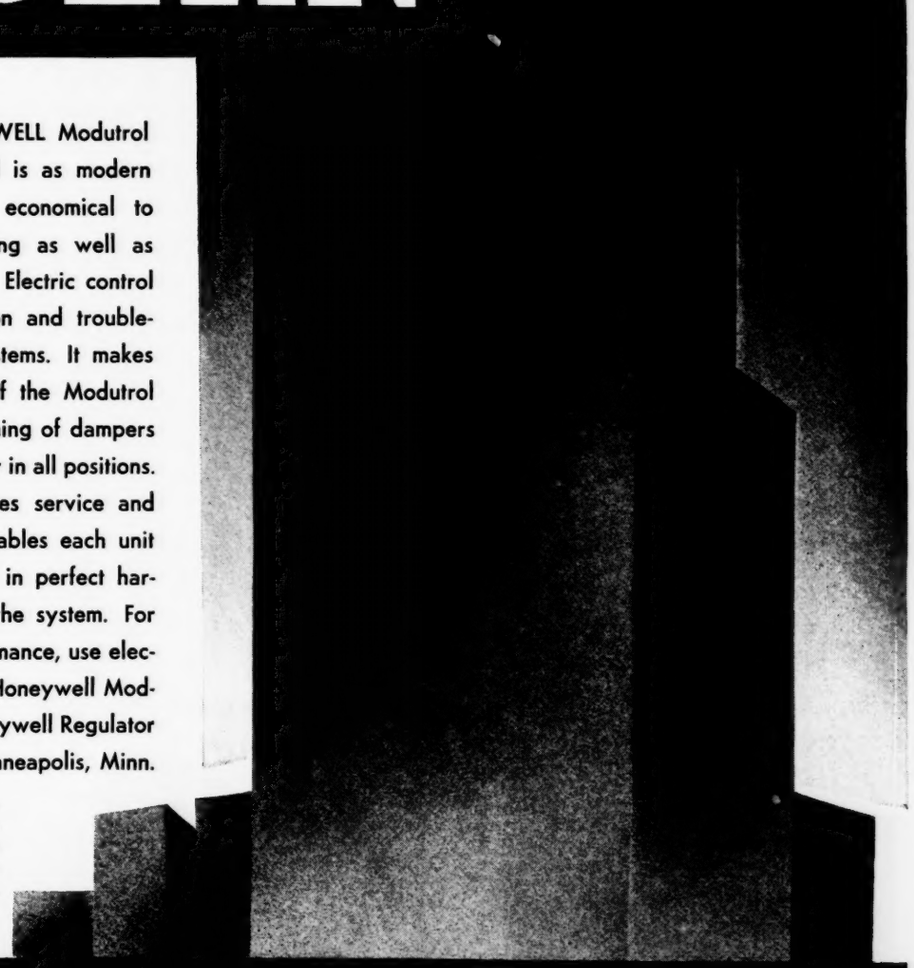
THE MINNEAPOLIS-HONEYWELL Modutrol System of electric control is as modern as electricity itself, and is economical to install and operate, in existing as well as new buildings—large or small. Electric control eliminates the costly installation and troublesome service of outmoded systems. It makes possible the true modulation of the Modutrol System with its precise positioning of dampers and valves, and with full power in all positions. Electric control greatly reduces service and maintenance costs, since it enables each unit to operate independently yet in perfect harmony with others comprising the system. For modern and economical performance, use electric control — the Minneapolis-Honeywell Modutrol System. Minneapolis-Honeywell Regulator Co., 2715 Fourth Ave. So., Minneapolis, Minn.

Branch and distributing offices in all principal cities . . . In Canada: Minneapolis-Honeywell Regulator Co., Ltd., 117 Peter St., Toronto. European sales and services: 233 Heerengracht, Amsterdam, Holland.



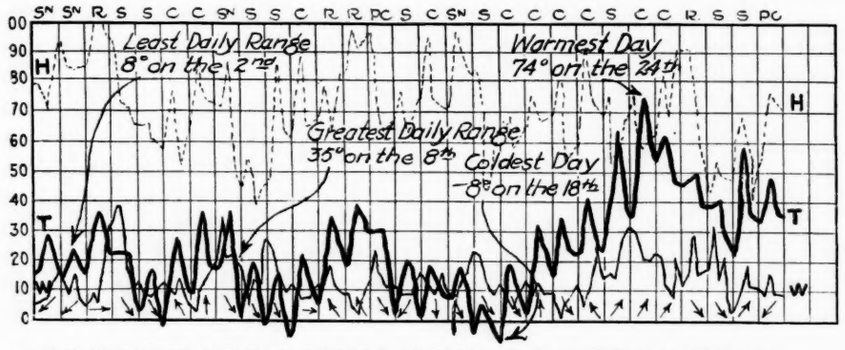
MINNEAPOLIS-HONEYWELL

Control Systems

BROWN INSTRUMENTS FOR INDICATING, RECORDING AND CONTROLLING

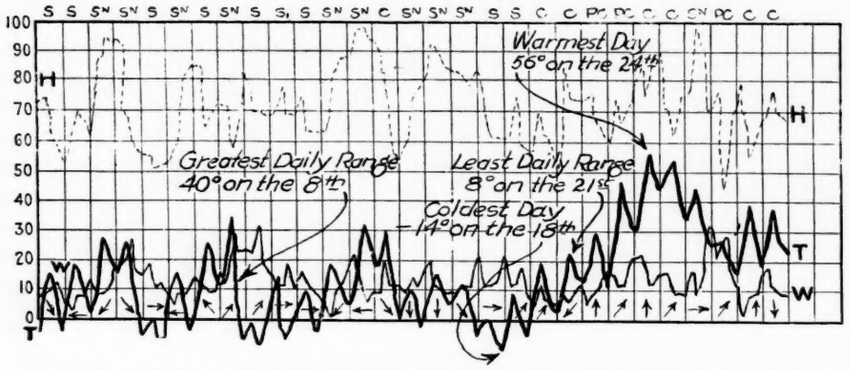
THE WEATHER FOR FEBRUARY, 1936

Plotted from records compiled for HEATING & VENTILATING by the U.S. Weather Bureau. Heavy curves (T), dry bulb temperatures in deg. F.; dotted lines (H), per cent relative humidity from readings at 8 a.m., noon, and 8 p.m. Light lines (W), wind velocity in m.p.h. Arrows indicate prevailing wind directions, north being arrow pointing up, etc.; S—clear; PC—partly cloudy; C—cloudy; R—rain; Sn—snow.



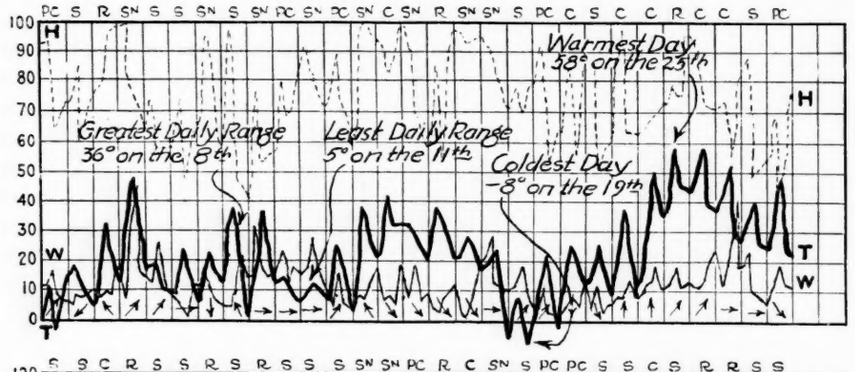
St. Louis

Mean temp. for month, 24.4°; aver. wind velocity, 11.8 m.p.h.; prevailing direction of wind, NW.



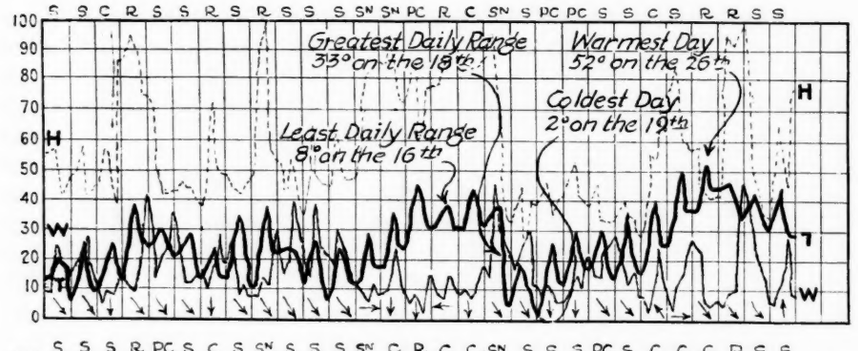
Chicago

Mean temp. for month, 15.2°; aver. wind velocity, 11.9 m.p.h.; prevailing direction of wind, W.



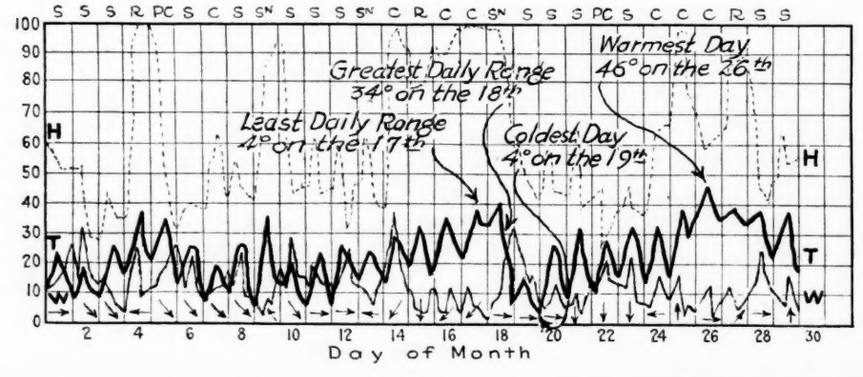
Pittsburgh

Mean temp. for month, 22.9°; aver. wind velocity, 11.2 m.p.h.; prevailing direction of wind, SW.



New York

Mean temp. for month, 26.6°; aver. wind velocity, 13.8 m.p.h.; prevailing direction of wind, NW.



Boston

Mean temp. for month, 22.8°; aver. wind velocity, 11.5 m.p.h.; prevailing direction of wind, W.

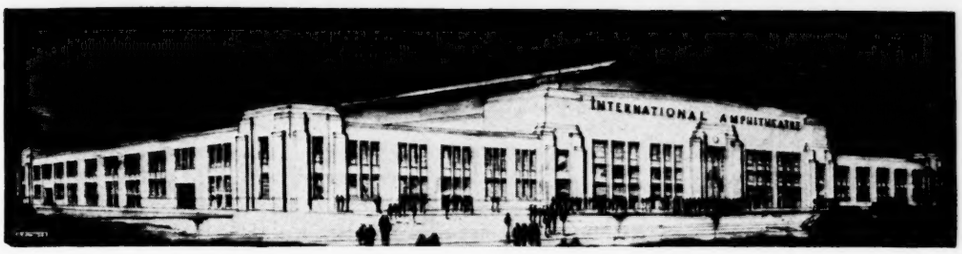
Pres. Chase Says: . . . . *"The finest conference and show program the industry has ever had."*

Mr. Frank D. Chase, Pres.,
Midwest Power Engineering Conference

The Power Event of 1936

POWER SHOW AND CONFERENCE

CHICAGO



APRIL
20-24th

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AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS
EDISON ELECTRICAL INSTITUTE
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MEET WITH THE MASTERS OF POWER

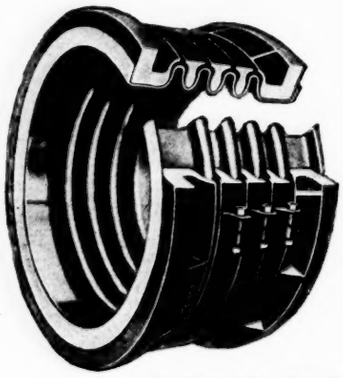
EXPANSION JOINTS

as manufactured by
E. B. BADGER & SONS CO.
BOSTON, MASS.

The Standard Corrugated Type Joint for Industry

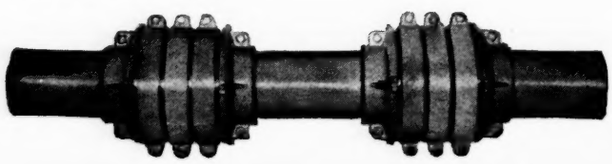
These expansion joints have back of them more than 40 years' experience. As made today, they embody many improvements brought out by Badger engineers from time to time. Two of the most recent are Directed Flexing and Special Controlled Heat Treatment, both of which have resulted in a better joint with longer life.

Be sure you buy or specify Expansion Joints made by E. B. Badger & Sons Co., Boston, Mass. You are then sure of getting joints long considered standard by engineers and operating men.



Standard Self-Equalizing Directed Flexing Badger Joint with flanged end. Can be fitted with Monel sleeves for protection against superheated steam.

Double welding end joints. Also made single. Welding end joints can be fitted with Monel sleeves.



BADGER EXPANSION JOINTS

Chicago Ventilating Contractors Claim Air Conditioning Work

(Continued from page 33)

unless based on or warranted by corresponding changes in plans or specifications, such changes being bid on by the original competitors only. . . .

Article IV

We further declare that by reason and virtue of the exacting requirements and responsibilities imposed upon the ventilating and air conditioning contractor, his domain is and as a matter of right ought to be the fabrication, erection and installation of mechanical ventilating and air conditioning systems as defined in Article V, and the purchase, erection and installation of all material and equipment necessary for such systems. Ventilating and air conditioning contractors, members of this Association, will reciprocate by a proper regard and observance of the functions of other mechanical trades in the building industry according to the agreed allocation of work.

Article V

The allocation of work is as follows:

To the ventilating and air conditioning contractors:

1—All sheet metal work with ducts, dampers, casings, etc.
2—Fans.
3—Motors and engines.
4—Controllers and starting equipment.
5—Air washers and their pumps and motors.
6—Filters and air cleaning devices.
7—Ozone machines.
8—Registers, grilles and mushrooms.
9—Air intakes and discharges.
10—Sound proofing.
11—Insulation of ducts and apparatus allocated to ventilating and air conditioning contractors.
12—Flexible connections.
13—Sheet metal work in connection with unit heaters.
14—Sheet metal work in connection with humidifiers.
15—Installation of dampers for automatic regulation.
16—Foundations for all ventilating apparatus.
17—Radiator enclosures and recess linings.
18—Temperature control, when electrical.
19—Furnishing and setting of any other apparatus or equipment, jurisdiction over which properly belongs to this trade.

Article VI

In cases where the entire installation cost is not of a considerable amount, it may be desirable for the contract to be awarded in its entirety. Under such circumstances, the ventilating and air conditioning contractor may become the principal contractor; or he may become a sub-contractor under others up to the sum of $2500.00. This sum must cover all equipment involved in the particular installation allotted to the ventilating and air conditioning contract, according to the allocation of work.

Article VII

Subject to all of the provisions of this Declaration

of Principles and Business Conduct, Ventilating and Air Conditioning Contractors will submit proposals for complete installations as defined in Article V as follows:

A—To Federal, State, County and Municipal Government Agencies, Public or Private Owners or Lessees of the buildings, structures, or properties in which the installations are to be made.

B—To qualified Architects and Engineers acting directly for Owners or Lessees. It is suggested that a contingent coordinating guarantee fund be provided in such cases where it may be deemed advisable.

C—To qualified General Building Contractors as direct representatives of Owners or Lessees where the Ventilating and Air Conditioning installation as defined in Article V is included in the general contract.

D—To Heating and/or Air Conditioning Contractors' Engineers qualified to promote and design air conditioning installations and who do assume entire responsibility for performance of guarantees as to temperature and humidities holding the Ventilating and Air Conditioning contractor responsible only for customary guarantees as to material and workmanship; provided that a like and equal reciprocal recognition is accorded to qualified Ventilating and Air Conditioning Contractors' Engineers; provided further, however, that in all cases where qualified Architects and Engineers are acting for Owners or Lessees, proposals by Ventilating and Air Conditioning Contractors shall be to such qualified Architects and Engineers.

E—Fair Trade Practice does not subordinate or subject ventilation and air conditioning work as defined in Article V to any trade or unnecessary credit hazard in the Building Construction Industry, except as limited, outlined and permitted under Article VI which reciprocal arrangement is suggested to facilitate handling of such small installations when desirable.

F—Fair Trade Practice requires that in all cases except as above limited, qualified and defined, ventilating and air conditioning installations as defined in Article V shall be treated as a separate and distinct trade in the Building Construction Industry.

G—The purpose of the above paragraphs is to remove handicaps from and encourage those engaging in the promotion of the heating and air conditioning industry, and not to remove emphasis from the desirability of Architects and Engineers issuing plans and specifications for installations separately for the various trades involved requiring no guarantees except for quality of materials and workmanship.

Article VIII

It will be considered at variance with the aforemade declaration of principles to submit a proposal on a complete installation where separate specifications have been issued.

• of Service Hot Water

Sylphon Thermostatic Water Mixers effectively limit the maximum temperature of hot water delivered to bathrooms, gang showers and kitchens, below the scalding point regardless of storage tank temperatures.

Utilizing hot water from any usual source, they regulate the amount of cold water required to temper the hot to the desired degree, actually mixing the two together before delivery.

Uneffected by sudden fluctuations in temperature or pressure of supply water, they assure accurate maintenance of safe temperatures, protect porcelain and plumbing fixtures. They not only save hot water but effect increased capacity of storage tanks by permitting water to be stored at higher temperatures.

The complete line ranging from sizes for individual showers to sizes for entire sections of large residential, commercial and industrial buildings, is described in Bulletin GH 40. Ask for it.

•

OTHER SYLPHON BULLETINS

Individual Room Temperature Control — Bulletins GH 80 and GH 70... Zone Control—Bulletin GH 70... Space Heat Control in Large Industrial Areas—Bulletins GH 50 and GH 70... Duct Type Heating and Air Conditioning Controls—Bulletin GH 50... Drinking Water Temperature Control — Bulletin GH 20.

FULTON SYLPHON CO.
KNOXVILLE. TENN., U.S.A.

Sales Representatives in Principal Cities

THE SUNDAY
DAVENPORT DEMOCRAT
CONVICT 3 IN SEA DISASTER
New Cold Wave Nears; Deaths Grow
PREDICT 15 TO 20 BELOW IN THE CITIES
OFFICERS OF DEATH SHIP FACE PRISON

Facsimile of front page of The Davenport Democrat

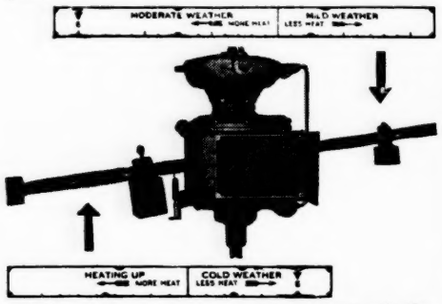
IOWA NEWSPAPER CUTS HEATING COSTS $517

Davenport Democrat Uses Webster Hylo System to Improve Heat Distribution, Lower Cost

Davenport, Ia.—The Democrat Building, home of the Democrat Publishing Company, publishers of the Davenport Democrat for more than 80 years, reduced the cost of heating during 1934-35 a total of $517, as the result of a Webster Heating Modernization Program which converted the installation into a Webster Hylo System of Steam Heating.

The modernization included necessary changes in piping to fit the installation to newspaper requirements, installation of radiator traps and accurately sized metering orifices in each of the 74 radiators and application of a central Webster Hylo Steam Variator Control.

The saving of $517 represents a 27 per cent reduction in cash expenditure for heating. At this rate, modernization will repay its entire cost in less than three years.



Illustrated at right is the Webster Hylo Steam Variator—a simple, rugged central control for use with Webster Systems where distribution of steam to all radiators is balanced by accurately sized metering orifices. The Hylo Variator Valve Operator (illustrated) automatically throttles a rugged motor-operated valve in the steam main.

There has been no "starving" of the heating system to effect these economies. As a matter of fact, the building is receiving substantially improved heating service with the Webster Hylo System.

Webster engineers cooperated closely with the owners and the heating contractors, the Ryan Plumbing and Heating Company, in meeting publishing requirements. The press room is kept at 80 degrees constantly. Savings are achieved without adverse effect on production.

If you are interested in (1) improved heating service and (2) lower heating cost in your building, address

Warren Webster & Company . . . Camden, N. J.
Pioneers of the Vacuum System of Steam Heating
Branches in 60 principal U. S. Cities . . . Darling Bros., Ltd., Montreal, Canada

1936 Business Off to Good Start as Building Contracts and Equipment Sales Rise

(Concluded from page 58)

for January, 1936, 444 were sold totalling 214,719 sq. ft. of heating surface, an increase of 134% in number and 70% in capacity. Cast iron round boilers showed a gain of 17%, while cast iron square boiler sales rose 30%.

Heat Distributing Equipment

In January, 1935, nearly 286,000 sq. ft. of various types of convectors were sold, as compared with nearly 574,000 sq. ft. in January, 1936, an increase of 101%. Cast iron direct radiators amounting to almost four million square feet were sold this January, as compared with slightly over three million square feet a year ago, an increase of 34%.

Building Contracts

Non-residential building in February was slightly over double that of the corresponding month in 1935. Residential building was almost double that of a year ago. The sharpest increase was that in hospitals and institutional buildings which rose 850%, but the most important increase (due to greater volume) was that in educational buildings, which increased 270%.

ABSTRACTS

(Continued from page 60)

to clinker at the temperature in the fuel bed.

To test this latter method a number of experiments were conducted at the Research Laboratory of the Battelle Memorial Institute, Columbus, Ohio, by Bituminous Coal Research, Inc. The tests were run in a small underfeed stoker, with a maximum feed of 30 lb. per hr., which was installed in a round boiler having a 22 in. firepot. The operation of the stoker was intermittent under a system of manual control following a definite schedule.

The loss of weight of the boiler was fixed to be 13.5 lb. per hr. which corresponds to a heat release of 193,000 B.t.u. per min. The hour was divided into three 20-min. cycles. The stoker was started at the beginning of one cycle and allowed to operate until one third the hourly loss, 4.5 lb., was burned; the stoker was then shut off until the beginning of the next 20-min. cycle and the operation repeated. Each test was 9 to 10 hr. in length.

The results of the tests are shown in the table. Although the ash of the coal used had a rather low softening temperature, 2,210°, the amount of clinker formed that could

RESULTS OF CLINKERING TESTS

Clinkering Material	Wt. Used Per Ton of Coal, Lb.	Cost of Treatment, Dollars Per Ton	Per Cent of Ash Formed Into Clinkers
None	—	—	12.3
Sodium Carbonate (Soda Ash)	20	0.52	8.3
Sodium Silicate (Water Glass)	42	0.50	11.5
Sodium Sulphate (Glauber's Salt)	88	0.55	12.7
Portland Cement	40	*	15.1
Borax	16.66	0.50	19.8
Sodium Metasilicate	40	1.42	27.7

*Not given.

be removed with tongs was not high. This was partly due to the short time of the test and partly to the relatively high excess air used. As the same conditions were maintained in the tests in which clinkering substances were used, the results should be comparable.

When sodium carbonate was used there were some signs of increased fusion of the ash but the per cent of clinker formed was actually less than with no treatment; no reason for this was apparent. Practically the same results were obtained with sodium silicate except that even less signs of increased fusion were seen.

With sodium sulphate the evidence of increased fusion was marked, even though the amount of clinker formed was practically the same as with no treatment. The Portland cement was only slightly effective.

Although there were no signs of general fusion over the clinkers when borax was used, some particles had been quite fluid as there were some sticking to the hearth. The percentage of clinkers was appreciably increased by the addition of borax.

The most effective treatment was the addition of sodium metasilicate. The clinkers obtained when this substance was used appeared more glassy than when the coal was untreated.

The tests showed that borax and sodium metasilicate were the most effective materials and are the logical materials to try in those emergency cases where adjustments of the stoker operating conditions are not effective and some action is imperative. It should be kept in mind that the results of these tests apply only for one coal and are not conclusive for other types.

[*"Clinker Formation in Small Underfeed Stokers," by Ralph A. Sherman and E. R. Kaiser. Information Bulletin No. 1, published by the Bituminous Coal Research, Inc., 803 Southern Building, Washington, D. C. Paper bound; 6 x 9 in.; 10 pages; price, 10 c.*]

NEW CATALOGS

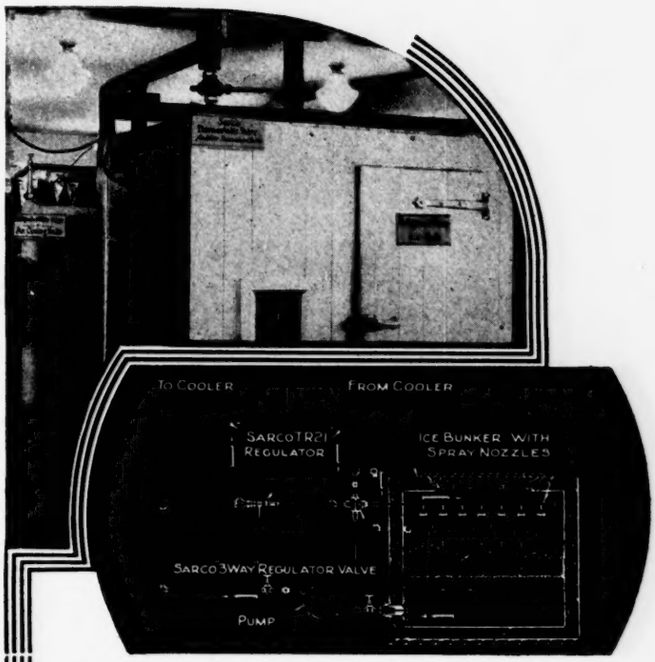
Buffalo Forge Co., Buffalo, N. Y. Bulletin 501, describing the company's type "PC" cabinets for air conditioning. Illustrated in color and with curves, engineering data, and examples showing how to determine the size of required cabinets. The data are arranged for both direct expansion cooling and for water cooling coils. Standard size, 24 pages.

Buffalo Forge Co., Buffalo, N. Y., has published a standard size, loose leaf catalog, Bulletin 2955, on its Buffalo Limit Load Conoidal fans. Illustrates and describes this equipment and includes performance curves and curves on sound records of these fans. Tables and dimensions complete the booklet; 16 pages.

A. M. Byers Co., Pittsburgh, Pa., has published two catalogs. The first, "Wrought Iron in Industry," is a 16-page, standard-size, well-illustrated bulletin showing the wide range of applications of wrought iron in industrial applications. The second is a special report prepared by the company's engineering service department and covers the use of wrought iron for flue gas conductors and coal handling equipment. Included with this material is a reprint of an article on the problem of the smokestack, suggestions for covering specifications, and design details for smokestacks. Standard size, 16 pages.

Ingersoll-Rand Co., New York. Form 1972-A, devoted to the company's condensate return motor pump. These units are available in either the one-pump or two-pump and in capacities ranging from 1,500 to 30,000 sq. ft. of radiator. Booklet describes and gives capacity data and specifications. Standard size, 8 pages.

United States Register Co., Battle Creek, Mich., has issued No. 25 Pocket Register Manual, a 3¾ x 6½ in., 72-page catalog of the company's line of registers and grilles.



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Ice manufacturers everywhere are giving increased attention to the development of air conditioning systems, using ice as the cooling medium.

It has been demonstrated that efficient cooling of restaurants, stores, bar rooms, theaters, etc., can be furnished at a very moderate cost, provided there is dependable automatic control.

Sarco has designed a special, self-operated Temperature Regulator for this purpose. It is attached to the ice cabinet as shown above and automatically by passes a portion of the return-water around the ice.

In this way only enough ice is used to maintain the desired temperature and waste is prevented.

Sarco Temperature Regulators operate by liquid expansion which gives them the same power at low temperatures as others develop only on heating service. They are ideal also for cold storage rooms, brine coolers, hot water heaters, drying rooms, cookers, pasteurizers, etc.

Write for catalog P-52.

SARCO COMPANY, Inc.
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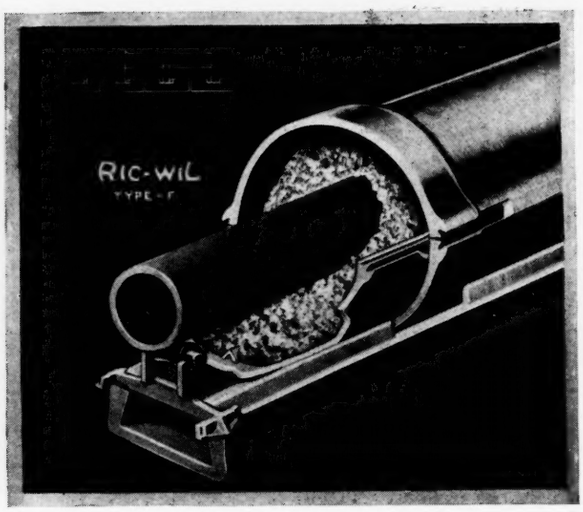
BRANCHES IN PRINCIPAL CITIES
SARCO CANADA LTD., FEDERAL BLDG., TORONTO, ONT., CANADA

SARCO TEMPERATURE REGULATORS



Why RIC-WIL Type "F" Conduit Means Dependable Protection for Underground Steam

- Famous Ric-wiL Dry-paC Waterproof Asbestos Insulation—for high efficiency. (Conductivity only .36 B.T.U. for low pressure.)
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- Famous Loc-liP side joint, continuous and visible.
- Best vitrified glazed tile—waterproof, everlasting.
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- Practical pipe supports which snap in place quickly.
- Only three pieces to handle for each 2 ft. of system, only three joints—low installation cost.

The above points emphasize the fact that Ric-wiL Conduit will keep *your* underground steam lines tight, dry, and 90% or more efficient. Ric-wiL Standard Systems include optional types of insulation and can be furnished in either tile or cast iron to meet any practical requirements as to supporting strength. Ric-wiL also manufactures the self-contained, pre-fabricated Unit Steam Main, a complete unit system for underground steam transmission. Catalog describing all types of Ric-wiL Systems gladly furnished on request.

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Two-Position

MASTER HEAT REGULATOR

A Railroad Pres'dent can't buy a better regulator than a "MASTER"—and here's a "MASTER" for the average home-owner. Noiseless, accurate, responsive to one degree of change, easy to install and will outlast the heating plant. Underwriters Laboratories list it as standard. A money-maker for you.

Made by the manufacturers of the famous Type B-144, the Original Gradual Control Heat Regulator.

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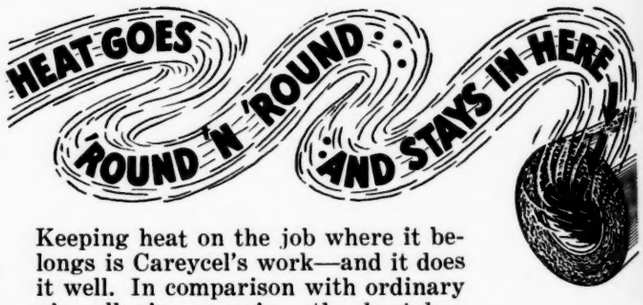

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Recommended for hot water and low pressure steam systems where temperatures range up to 300° F.

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